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AN EXPERIMENTAL RINGBOM-STIRLING ENGINE

by

G. Walker and O.R. Fauvel
Department of Mechanical Engineering
University of Calgary
Calgary, Alberta

for the
Transportation Development Centre
Transport Canada

Report No. TP-6596E Aug. 1985

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17. Abstract

A proof-of-principle Ringbom-Stirling engine built using parts from a commercially available diesel engine has been demonstrated. The engine used air as the working fluid and water as the lubricant. It was developed as part of a wider study concerned with large coal-burning Stirling engines for use in railway locomotives, ships, off-highway vehicles for mining, construction, agriculture, forestry and for stationary power.

Conversion of diesel engines to Ringbom-Stirling engines offers the prospect of gradual conversion to coal-burning operation rather than direct replacement by new equipment.

The experimental engine was not tested under the design conditions of speed and pressure because of safety concerns. The results achieved at low pressures and speeds indicate that design output would probably be achieved at the design conditions.

A computer simulation program developed during the work was used to perform parametric studies of the effects of change in the principal design parameters. This study revealed the significance of the mass of the displacer in the proper operation of a Ringbom engine including speed control. An important consequence was the concept for a new type of Ringbom-Stirling engine with independent displacer drive.

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17. Resume

Un moteur Ringbom-Stirling devant servir à démontrer le principe en jeu, et réalisé en modifiant un moteur diesel de fabrication courante, a été essayé; il a utilisé l'air comme fluide de travail et l'eau comme lubrifiant. Sa réalisation fait partie d'une étude s'intéressant aux gros moteurs à cycle de Stirling chauffés au charbon et destinés à la propulsion de locomotives, de navires, de véhicules hors routes, ainsi qu'à la production d'énergie et à diverses autres applications minières, agricoles, forestières, etc.

Le cycle Ringbom-Stirling ouvre la perspective d'une conversion graduelle des moteurs actuels à combustion interne en moteurs à combustion externe brûlant du charbon, évitant ainsi leur remplacement intégral.

Ce moteur expérimental n'a pas été essayé dans les conditions normales de vitesses et de pressions, pour des raisons de sécurité. Mais les résultats ont montré que le rendement attendu aurait pu être atteint si les vitesses et les pressions mises en oeuvre avaient été normales, au lieu d'être réduites comme cela fut le cas.

Un programme de simulation assistée par ordinateur, mis au point en cours de route, a été utilisé pour vérifier l'effet des variations dans les paramètres de base. C'est ainsi qu'a été mise en évidence l'importance de la masse du déplaceur dans le rendement d'un moteur à vitesse variable. Ces travaux ont débouché sur un modèle nouveau de moteur Ringbom-Stirling disposant d'un entraînement séparé pour le déplaceur.

18. Mots-clés	19. Diffusion
Moteur à cycle de Stirling. Moteur Ringbom de traction ferroviaire. Charbon.	Le CDT dispose d'un nombre limité d'exemplaires.

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Professor James Senft of the Department of Mathematics and Computing Science, University of Wisconsin, River Falls, Wisconsin, and formerly at Sunpower Inc., was a consultant on theoretical aspects of Ringbom-Stirling operation. He contributed substantially to the understanding of the 'over-driven' mode of operation of Ringbom-Stirling engines.

Dr. William Martini, Stirling Engine Consultant, Richland, Washington contributed significantly to the development of the computer simulation programme for Ringbom-Stirling engines.

Staff of the University of Calgary participating in the programme and contributing significantly to it included:

Dr. O.R. Fauvel, Dr. J.A.C. Kentfield, Mr. V. Srinivasan, Mr. J. van Benthem, Mr. R. Gustafson, Mrs. K. Undseth, Mr. N. Vogt, Mr. B. Unterberger, Mr. W. Eremko, Mr. B. Wilkinson, Mr. W. Anson and Mr. A. Moehrle.

TABLE OF CONTENTS

1.	INTRODUCTION	1
2.	COMPUTER SIMULATION PROGRAM	5
2.1	The Martini Isothermal Second Order Analysis	5
2.2	Crank-Driven Piston Displacer Simulation	6
2.3	The TDC Ringbom-Stirling Engine Simulation	6
3.	ENGINE DESIGN	12
3.1	General	12
3.2	Seals	16
3.2.1	Displacer Seal/Guide Rings	20
3.2.2	Displacer Rod Seal/Guide Bushing	20
3.2.3	Gas Spring Piston Seal	20
3.2.4	Piston Seal/Guide Rings	20
3.2.5	Piston Rod Seal	21
3.2.6	Static Seals	21
3.3	Bearings and Lubrication	21
3.4	Flywheel Design and Engine Balancing	23
3.5	Firing Equipment	24
4.	ENGINE SPECIFICATIONS	25
4.1	Components	25
4.1.1	Crankcase	25
4.1.2	Crankshaft	25
4.1.3	Crankshaft Bearings	25
4.1.4	Crosshead Bearing	26
4.1.5	Crosshead	26
4.1.6	Piston Rod	26
4.1.7	Piston Rod Seal	26
4.1.8	Piston	26
4.1.9	Displacer	26
4.1.10	Heater	27
4.1.11	Regenerator	27
4.1.12	Cooler	27
4.1.13	Gas Spring	27
4.2	Operating Conditions	28
4.2.1	Speed	28
4.2.2	Temperature - Heater	28
4.2.3	Temperature - Cooler	28
4.2.4	Pressure	29
5.	INSTRUMENTATION	30
5.1	MACSYM II System	30
5.2	Measurement of Engine Operating Parameters	30
5.2.1	Engine Speed	30
5.2.2	Engine Torque	31
5.2.3	Piston Position	31
5.2.4	Displacer Position	31

5.2.5 Working Space Gas Pressure 31
5.2.6 Balance Space Gas Pressure 32
5.2.7 Gas Spring Space Gas Pressure 32
5.2.8 Heater Tube Temperatures 32
5.2.9 Heater Head Temperature 32
5.2.10 Regenerator Housing Temperature 32
5.2.11 Coolant Water Flow and Temperature 32
5.2.12 Lubricating Water Supply Pressures 33

6. PRE-TEST TRIALS 34
6.1 General 34
6.2 Operating Data 34
6.2.1 Chart Output 35
6.3 Analysis of Operating Data 35

7. COMMENTARY AND CONCLUSIONS 45

REFERENCES 54

APPENDIX A - BASIC CHECKLIST FOR ENGINE START-UP AND SHUT-DOWN 55

LIST OF FIGURES

Figure 1	Performance Characteristics of the Free-Displacer TDC Ringbom-Stirling Engine	9
Figure 2	Effect of Displacer Rod Diameter on Engine Power Output and Efficiency of Free Displacer TDC Ringbom-Stirling Engine	10
Figure 3	Effect of Displacer Mass on Engine Power Output and Efficiency of Free-Displacer TDC Ringbom-Stirling Engine	11
Figure 4	Lister Model 8/1 Diesel Engine	13
Figure 5	Ringbom-Stirling Engine	14
Figure 6	Cross-Section of the TDC Ringbom-Stirling Engine	15
Figure 7	Cooler Subassembly	17
Figure 8	Heater Head	18
Figure 9	Inverted Heater Dome Showing Regenerator with Retaining Ring and Woven Matrix Material	19
Figure 10	Sample of MACSYM Data Output, Part I	41
Figure 11	Sample of MACSYM Data Output, Part II	42
Figure 12	Recorded Motion of Piston and Displacer for Three Different Tests. Displacer motion (upper) and piston motion (lower) are in the same senses as the traces with TDC being uppermost.	43
Figure 13	Measured Power Output as a Function of Engine Speed and Mean Pressure	44
Figure 14	Concept for Bingham-Stirling Marine Engine	50
Figure 15	Concept for Siemens-Stirling Marine Engine	51

LIST OF TABLES

Table 1	Summary of Experimental Results, Part I	38
	Summary of Experimental Results, Part II	39
	Summary of Experimental Results, Part III	40

1. INTRODUCTION

A Stirling engine is a heat engine absorbing heat at high temperature from some external source, converting some to work, and rejecting the balance as waste heat near ambient temperatures.

High temperature heat is supplied to the engine from an external source, and can be of any origin, combustion of solid and liquid fuels, concentrated solar energy, or radioisotope heat. In this regard Stirling engines are similar to Rankine cycle engines (steam engines). Both are candidates for coal-burning engines as alternatives to oil-burning diesel engines.

The principal difference between Rankine and Stirling engines is in the thermodynamics of the cycles. The Rankine cycle involves the evaporation and condensation of the working fluid and so the maximum and minimum temperatures of the cycle are intimately related to the maximum and minimum pressures in a way characteristic of the working fluid. The result is an ensemble of machinery, boiler, condenser, reciprocating engine or expansion turbine and auxiliaries that is large, heavy but with a high thermodynamic efficiency, i.e. 40%. Systems which operate without a condenser are smaller and lighter but have a low thermodynamic efficiency, i.e. 10%.

A Stirling engine uses a gaseous working fluid and therefore the pressure and temperature levels may be independently chosen. Moreover, the thermodynamic cycle incorporates a regenerative heat exchanger which 'recycles' a substantial fraction of the energy flowing through the system. As a consequence the Stirling engine has the potential to be a coal-burning unit of moderate size and weight and with a reasonable thermodynamic efficiency, i.e. 30/35%.

Both Stirling and Rankine engines can operate on liquid fuels but neither offer any economic or performance advantage over internal-combustion engines; so long as diesel oil is available, the best engine for its use is the diesel engine. Nevertheless, it is prudent to recognise that coal is the world's most abundant fossil fuel and to consider, for future application, the alternatives to oil-burning engines.

In this spirit of enquiry Walker (1981a)* examined the possibilities of the Stirling engine for railway locomotive application with particular regard to the Canadian environment. This report is an important reference document to the topic of interest here and was well summarized by Walker *et al* (1982).

In the locomotive application study the proposal was made for

*References given in full at the end of the text.

novel Ringbom-Stirling engines using air as the working fluid and water as the lubricant. It was further proposed that such engines could be made by converting existing diesel locomotive engines using many of the same parts and components and that savings could therefore result from the gradual conversion of existing diesel-powered locomotive fleets to coal-burning operation, rather than the outright replacement required for steam locomotives. The modified diesel engine working as a Ringbom-Stirling unit would operate in combination with a fluidised bed coal combustor and a sodium heat pipe thermal transport system.

The report further recommended the concept be investigated, by a proof-of-principle demonstration Ringbom-Stirling engine converted from an industrial diesel engine. That investigation is the substance of the report presented herein.

The essential elements of a Stirling engine include two spaces, at different temperatures, whose volumes can be varied cyclically but out of phase by approximately 90°. In a power system the hot space is called the expansion space and the cold volume is the compression space. The two spaces are coupled through three heat exchangers, the heater, regenerator and cooler.

A Ringbom-Stirling engine is a special form of Stirling engine characterized by a free displacer and a piston coupled by a connecting rod to the engine crankshaft. The displacer moves in the displacer cylinder entirely under the action of pressure forces upon it. The specialist technology of free-piston engines has been summarized by Walker *et al* (1985).

Engines of this type were invented at the University of Calgary in 1970 (see Walker (1973)) and were called 'Hybrid-Stirling engines'. The principal attraction was the potential offered to convert internal combustion engines to Stirling operation. Later the original 1907 patent of Ossian Ringbom became known and Stirling engines with free displacers were then called Ringbom-Stirling engines.

Little previous work on Ringbom-Stirling engines has been reported in the literature but a 1 horsepower prototype unit was developed in 1980/81 by Sunpower Inc., Athens, Ohio in a program sponsored by a major United States engine builder. Sunpower Inc. is a research/development company specialising in free-piston Stirling engine systems. The 1 horsepower prototype engine operated successfully but was considered by the sponsor to be uneconomic for commercial production. For this reason the prototype development was not continued beyond the initial stage.

A participant in this pioneer Sunpower development, Dr. J. Senft, now a professor at the University of Wisconsin, was engaged as a consultant for the TDC/University of Calgary Ringbom-Stirling engine project. His report, Senft (1981), and subsequent publications (Senft (1981), Senft (1983)) are the pioneering contributions to the theory of Ringbom-Stirling engines.

Following a careful evaluation of commercial diesel engines the Lister 8/1 single-cylinder slow speed diesel engine was selected for conversion. An engine of this type was obtained and the engineering design for the conversion was carried out. Complete details of the design study were presented in a design report (Walker (1981b)).

Construction of a Ringbom-Stirling engine using parts of the Lister engine and the creation of an engine test bed with the associated instrumentation was carried out at the University of Calgary and the unit was installed in temporary facilities. Engine development and testing commenced shortly thereafter.

Considerable developmental work was carried out to bring the engine to an operating condition. Following this some 20 hours of actual test time was accomplished on a schedule of increasing pressure and speed. The maximum power achieved was a net engine output of 1.85 KW at a mean pressure and speed of 2.5 MPa (371 lb per sq in) and 6.9 Hz (424 R.P.M.). Operation of the engine at higher pressures and speeds in reasonable safety was not possible in the temporary installation and it was decided to cease the further engine tests to raise the operating conditions and power output to the design point.

A prototype Ringbom-Stirling engine with bore and stroke characteristics identical to the Lister diesel engine employing air as the working fluid and water as the lubricant was built and operated to produce a surplus of power at the shaft. The further work necessary on the engine was thought to be largely of an engineering development nature best carried out in an establishment with adequate test facilities, primarily an engine test cell.

The objective of the further work required would be to increase the engine power output by increase in speed and pressure levels, to simplify and improve the engine control and combustion systems, to investigate alternative corrosion-resistant coatings and to optimise the water lubrication/seal/bearings system from the aspect of design and operation.

The project is regarded as successful overall despite the limitation on speed and pressure levels due to safety considerations. In course of the work the fundamental theory of Ringbom-Stirling engines was developed and verified experimentally, particularly with regard to displacer motion. A small model Ringbom-Stirling engine constructed independently of the project in conjunction with high speed photographs confirmed the actual motion of the displacer was virtually identical to that predicted for Ringbom-Stirling engines in the consultant's report by Dr. J. Senft.

A sophisticated computer simulation program for Ringbom-Stirling engines was developed in course of the project and was used to establish a rational basis for the design of Ringbom-Stirling engines.

Finally, several original design concepts for major components of Stirling engines emerged during the project and will be incorporated in future engine developments.

For the period January to July 1983 the Principal Investigator was on leave from the University of Calgary to study the technology of sodium heat pipes and their application to Stirling engines. A report (Walker (1983)) summarizes the findings. Although not part of the work reported here it is an important reference directly relevant to the combination of a Stirling engine system with a fluidized bed coal combustor.

In summary, at this juncture the experimental engine has not been subjected to sufficient testing to establish its potential operation performance. Continuation of testing at design pressures and speeds has been postponed until the U of C acquires adequate test facilities or TDC decides to resume development and testing elsewhere.

2. COMPUTER SIMULATION PROGRAM

As part of this project considerable effort was invested in the development of a computer simulation program for Ringbom-Stirling engines. This work was described by Srinivasan (1982) in a T.D.C. report, more extensively Srinivasan (1983), with the actual program listing and further results, and in summary form by Srinivasan *et al* (1984).

A commercially available second-order isothermal Stirling engine simulation program (Martini (1981)) was modified for operation on a Commodore Pet Micro-computer. In the existing form the program was arranged for both single and double-acting dual piston Stirling engines with air, helium or hydrogen as the working medium. Sinusoidal piston and displacer motion was assumed.

Modifications and extensions to the program were introduced to facilitate simulation of the Ringbom-Stirling engine, a distinctive class of Stirling engine characterized by a free displacer and crank-controlled piston.

2.1 The Martini Isothermal Second Order Analysis

The basic assumptions and extensive discussion for the elementary programme has been given by Martini (1981). The program allows any motion of the pistons or piston and displacer to be specified for computation of cold space and hot space volumes. Initially, the hot space temperature is set at the hot metal temperature and the cold space temperature at the inlet cooling water temperature. The regenerator is assumed to be at the log mean temperature of these two temperatures. These values are modified in the course of the calculations.

At each incremental value of the crank angle the instantaneous hot and cold space volumes are calculated and therefrom the instantaneous pressure, assuming the ideal gas law $PV = M(R)T$. The pressure vs. total volume curve is integrated numerically to yield the energy output per cycle and integration of the pressure-expansion space curve yields the heat input per cycle. An approximation is made for the cycle time the gas enters and leaves the hot and cold spaces based on mass fraction data in the hot and cold spaces. The flow time through the regenerator is then assumed to be the average of the hot and cold flow times. Pumping flow losses in the heater, cooler and regenerator are now computed and subtracted from the basic power to give the indicated power. A mechanical efficiency for the engine is assumed and friction power is deducted from the indicated power to give the brake power of the engine.

The various heat losses, reheat, shuttle, pumping, temperature swing and conduction losses are computed and added to the basic heat input. Then because flow friction creates heat, the flow loss through the heater and half the flow loss through the regenerator is deducted

from the above to obtain the net heat input to the heater.

The temperature rise in the cooling water is calculated from the specified inlet cooling water temperature and flow rate, and the cold metal temperature, initially set equal to the inlet cooling water temperature, is now corrected. The heat transfer coefficients in the heater and cooler are estimated and a comparison made of the gas cooler load and the theoretical heat transfer due to the temperature difference and mass flow. A similar comparison is made for the heater. Subsequently these gas temperatures are adjusted, i.e. the hot space temperature is decreased while the cold space temperature is increased. Convergence criteria are satisfied when the hot and cold gas temperatures are such that the computed heat transfers in the heater and cooler are equal to the sum of heater and cooler loads for the basic engine operating between the effective hot and cold space temperatures plus all losses.

This isothermal second-order analysis was modified and extended for the TDC engine simulation, both for a crank driven piston and displacer and for a free displacer Ringbom-Stirling arrangement.

2.2 Crank-Driven Piston Displacer Simulation

The Martini programme was run on the Commodore Pet microcomputer using the actual dimensions of the TDC engine but assuming the engine to have kinematic-drive with sinusoidal piston and displacer motion. To calculate the volumes above the piston, below the displacer and above the displacer, arrays were defined for the various working space volumes. Corrections for the friction factor for the regenerator matrix were incorporated in the model.

In the Martini analysis, the mass fractions at the different points of the engine are calculated for 360/ND points during the cycle (ND is the selected crank rotation increment) and the gas flow is assumed to comprise alternate periods of constant flow and zero flow. As the mass fractions in the working spaces varies with phase angle, an approximation of the flow time corresponding to that phase angle was introduced.

Simulation of the TDC engine was carried out for a constant speed of 16.6 Hz (1000 rpm) with different phase angles and for different speeds with a phase angle of 90°.

Little variation in brake power output was found at different phase angles. The basic cycle power increased with phase angle from 50° to 90° but this was largely offset by an increase in the aerodynamic flow losses arising from the higher fluid velocities. The indicated and overall efficiencies increased with lower phase angles indicating that relatively low phase angles of 60°-70° will result in lower pumping losses and some gain in thermal efficiency.

2.3 The TDC Ringbom-Stirling Engine Simulation

Generation of the pressure-time data for a crank-driven engine

facilitated computation for the Ringbom-Stirling engine of the free displacer motion under the influence of the pressure differential acting over the area of the displacer rod. Initially, the piston and displacer were assumed to be crank-driven at any given phase angle, (e.g. 90°) and the instantaneous pressure was calculated as described earlier. Convergence criteria are set at this stage requiring the pressures at the beginning and end of the cycle to be within 0.01 percent of each other and the mean pressure to be within one percent of the specified mean gas pressure.

Using the generated pressure-time curve, the displacer motion was derived in terms of the changing workspace gas pressure acting on the area of the displacer rod. By designing to utilize this force, Senft (1981) introduced the concept of "overdriven mode operation". This was assumed to occur when the displacer is driven by the changing workspace gas pressure in such a way that the displacer reaches its physical travel limit before reversal of the gas pressure force acting across the rod takes place.

Senft (1981) assumed the displacer to be at the hot end with the piston at midstroke at time t = 0. At that instant, the pressure difference across the displacer rod acted to hold the displacer at the hot end. As the piston rose, a point was reached at time t_0 where the pressure difference across the displacer rod area was zero. As the piston moves beyond this point, the pressure in the working space increases to exceed the pressure in the spring space (assumed to be constant). The displacer is then accelerated toward the cold end. The displacer reaches the limit of its stroke at time t_1 and awaits the commencement of its return stroke beginning at a time t_2, exactly half a period from t_0. This sequence, described by Senft, was used here to derive the displacer motion from a given pressure-time curve, calculated initially for a displacer with sinusoidal motion and corrected in subsequent iterations.

To start, at time t = 0, the velocity XV = 0 and the displacer is at its maximum rise position. The displacement and velocity for the first time interval were calculated and the process repeated for the whole cycle with initial conditions taken from the previous values of velocity and displacement.

The hot and cold space volumes were recalculated for the new displacer and piston positions and therefrom the pressure variation in the engine. The process of evaluating a fresh set of displacer positions from the pressure function continued until the convergence criteria were satisfied. Finally the thermal and flow losses, the power output and thermal efficiency were calculated as before.

Estimates of the fraction of cycle time during which gas entered and left the hot and cold spaces (FH,FC) were derived from a study of the mass fractions of gas in hot and cold spaces at different speeds. Values of the cycle times FH and FC were entered as functions of engine speed as program input.

To expedite the project the principal design and operating

parameters of the prototype engine were initially defined in arbitrary fashion and the actual mechanical design of the engine was carried out, drawings of the parts prepared, and manufacturing of the parts started before the computer simulation program became available. This did not cause the problem it might appear to have done. It was anticipated that, as is normally the case, the computer program would need modification in the light of actual test bed results to predict engine operation with reasonable accuracy. Thus the computer program was viewed as supplementary to the experimental engine, principally for subsequent explorations of the effects of parameter variation, for optimization of the prototype design and for the mechanical design of larger, more complicated multiple-cylinder Ringbom-Stirling engines.

Parametric studies of the prototype engine were carried out using the simulation program to investigate the effect of modifications to the principal design parameters. Figure 1 shows typical results calculated for the change in engine power output and efficiency as a function of speed for given operating conditions of high and low cycle temperatures and mean pressure. Figure 2 shows the effect of the displacer rod diameter and Figure 3 the effects of variation in the displacer mass.

Under the simulated conditions of the overdriven mode operation, it is evident that:

1. As rod diameter increases at fixed displacer mass and engine speed, the basic power and brake power increase with a slight reduction in efficiency.

2. As displacer mass is increased for a given rod diameter and engine speed, the basic power and brake power decrease with a slight increase in efficiency. These trends are in accord with the theoretical inference of Senft (1981).

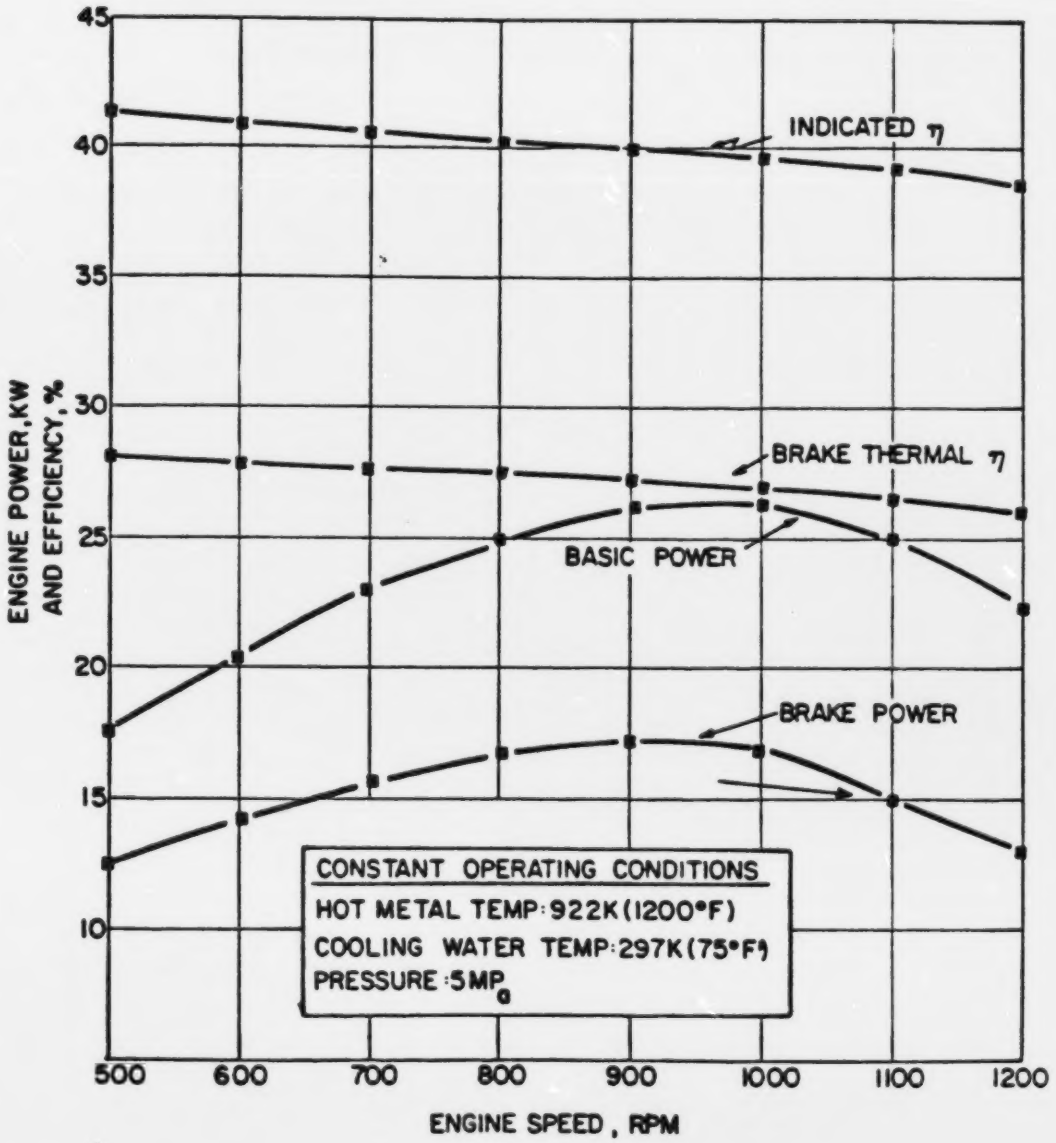

FIGURE 1 PERFORMANCE CHARACTERISTICS OF THE FREE-DISPLACER TDC RINGBOM-STIRLING ENGINE

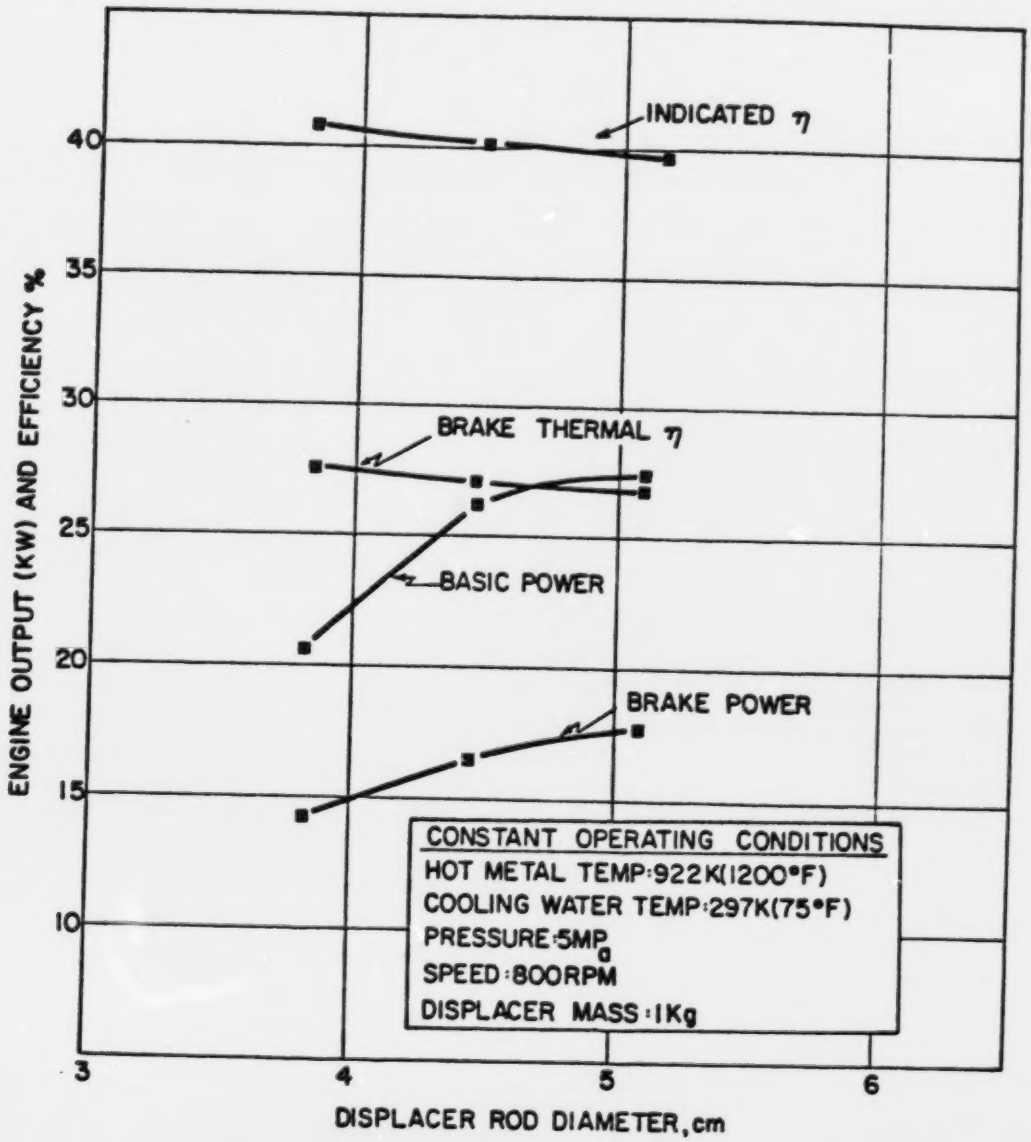

FIGURE 2 EFFECT OF DISPLACER ROD DIAMETER ON ENGINE POWER OUTPUT AND EFFICIENCY OF FREE DISPLACER T.D.C. RINGBOM-STIRLING ENGINE

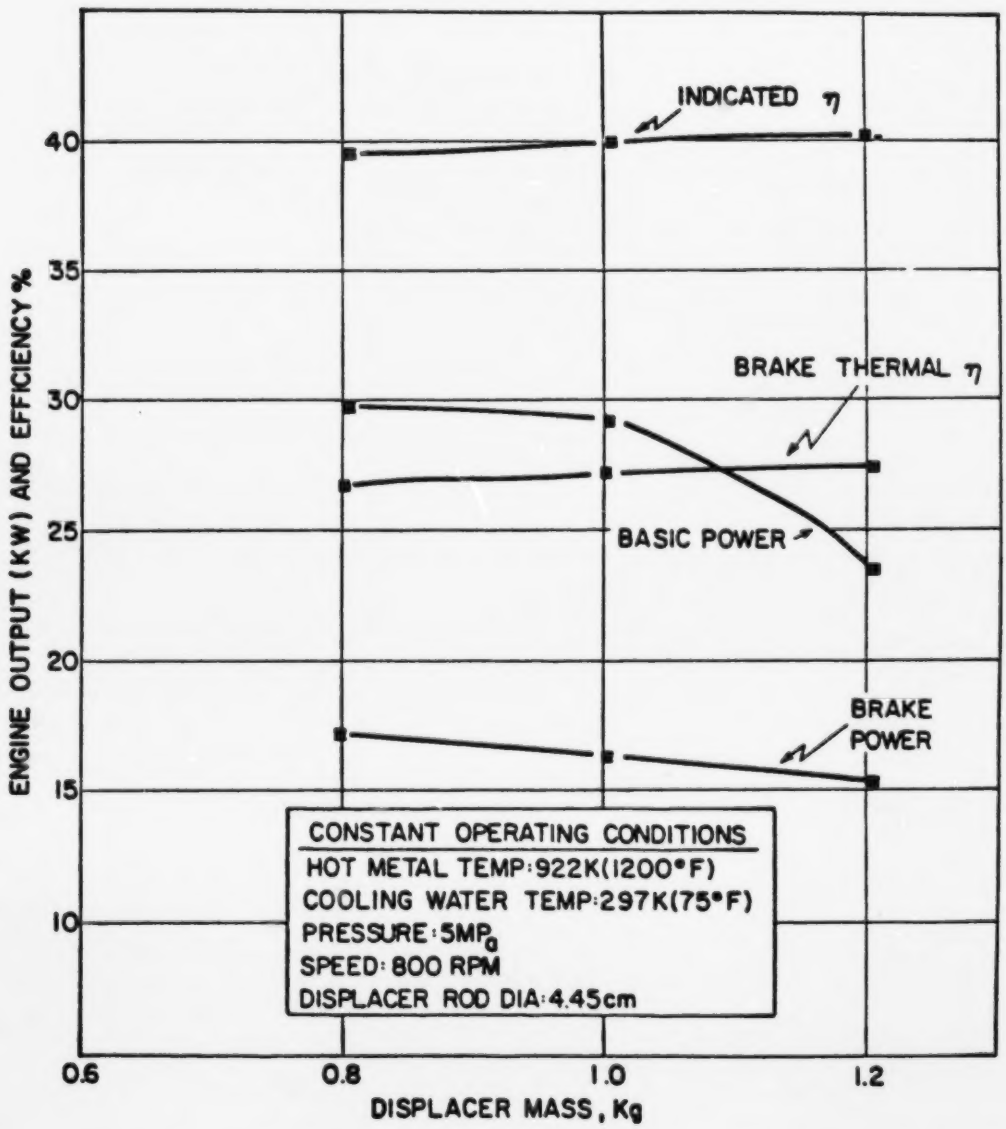

FIGURE 3 EFFECT OF DISPLACER MASS ON ENGINE POWER OUTPUT AND EFFICIENCY OF FREE DISPLACER T.D.C. RINGBOM-STIRLING ENGINE

3. ENGINE DESIGN

3.1 General

The mechanical design of the prototype engine was carried out as the initial task of the project well ahead of the development of a computer simulation program. The design was accomplished using the best available methods to fulfill an engine specification arbitrarily selected.

Sunpower Inc. were retained as design consultants for the engine design process. Drs. Walker and Fauvel visited Sunpower during the design phase and were generously afforded access to the Sunpower computer simulation programs to facilitate the design/decision making process. Sunpower staff engineers G. Wood and D. Gedeon along with the president W. Beale were principally involved with Walker and Fauvel in the design.

General arrangement and detail component drawings for the complete engine system and the test bed installation were prepared and submitted to TDC for approval. The original engineering drawings are on file in the Department of Mechanical Engineering, University of Calgary.

The design report (Walker (1981b)) contains a full account of the detailed design considerations and associated work.

The Lister 8-1 engine shown in Figure 4 was used to provide the crank, crank case and flywheel of the experimental engine. The Lister cylinder was used as a crosshead guide. The Lister engine is rated at 5.9 KW (eight horsepower) with a bore of 114.3 mm (4.5 ins) and a stroke of 133.35 mm (5.25 ins) running at 13.33 Hz (800 revolutions per minute). The engine is conservatively rated, of vintage design and principally used for agricultural purposes as a small stationary power plant. It was well-suited for this project typifying in many respects railway diesel engine technology, i.e. slow running, conservative rating. Many parts were removed (cylinder head, fuel pump, etc.) and substantial mass was added to the twin flywheel.

The engine was completely disassembled and all the internal surfaces plated with poly-tetra-fluoro-ethylene, as a protection against corrosion by the water lubricant. The journal surfaces of the crankshaft were also chrome-plated for corrosion protection.

The experimental Ringbom-Stirling engine is shown in Figure 5 and in cross-section Figure 6. The piston of the diesel engine was replaced with a crosshead and was coupled by a piston rod to the power piston of the Stirling engine. The space *above* the power piston was the 'compression space' of the Stirling engine.

This space was in communication with a second cylinder containing the displacer, a hollow vessel consisting of a stainless steel dome on

Figure 4 Lister Model 8/1 Diesel Engine

Figure 5 Ringbom-Stirling Engine
Produced from Lister
Diesel Engine

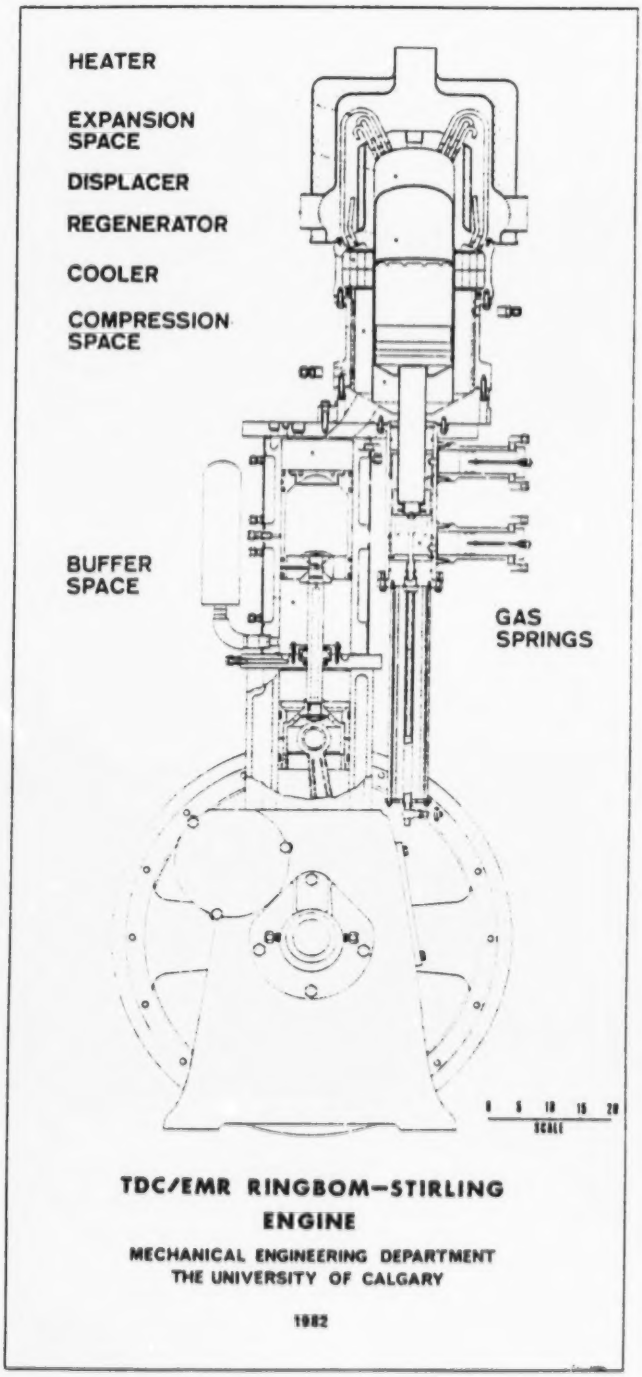

Figure 6 Cross-Section of the TDC Ringbom-Stirling Engine

a stainless steel base.

The displacer cylinder comprised an array of parts consisting of a water cooled 'cooler' (Figure 7) at the bottom and a thick wall stainless steel 'heater' at the top (Figure 8). A close packed wire 'regenerator' was sandwiched between (Figure 9).

The space above the displacer was the 'expansion space'. The space below the displacer was part of the compression space in direct communication with the compression space above the piston.

The expansion space was connected to the dual compression spaces through internal flow passages in the cooler, void spaces in the regenerator and an ensemble of heater tubes shown in Figure 8.

A displacer rod of relatively large diameter was attached to the displacer with a spring piston fitted to it at the lower end. The piston operated in a spring cylinder. Gas springs of variable volume were provided above and below the spring piston.

The displacer is a low mass structure caused to move in the displacer cylinder by the difference in pressure between the expansion space and the spring cylinder acting over the area of the displacer rod. The gas springs serve to decelerate the piston at the end of each stroke and to help reverse the direction of motion of the displacer.

The heater head was enclosed in a thermally insulated dome equipped with inlet and exit ports. Hot combustion gases produced by the combustion of propane and oxygen entered through the inlet port, heated the heater tubes and then exited the dome. The gases were burned in two standard commercial heating torches in a combustion chamber immediately upstream of the heater dome.

The space below the power piston is called on Figure 6 the 'buffer space'. This was pressurized to the mean cycle pressure in the working space so as to relieve the load on the running gear of the engine. Pressurization of the buffer space to $\bar{p}$ (the mean cycle pressure) reduces the pressure difference across the piston to $(p_{ins}-\bar{p})$ rather than $(p_{ins}-p_{at})$ with an unpressurized buffer space. p_{ins} is the instantaneous pressure in the engine working space, p_{at} is the atmospheric pressure.

In the fully developed double-acting reservoir of the engine the buffer space would become a second compression space acting in concert with another displacer cylinder to form a second Stirling engine system. On grounds of economy and to expedite the project the second Stirling engine system was never incorporated in the prototype engine.

3.2 Seals

Various dynamic and static gas seals were provided in the engine. Critical dynamic seals were located on the:

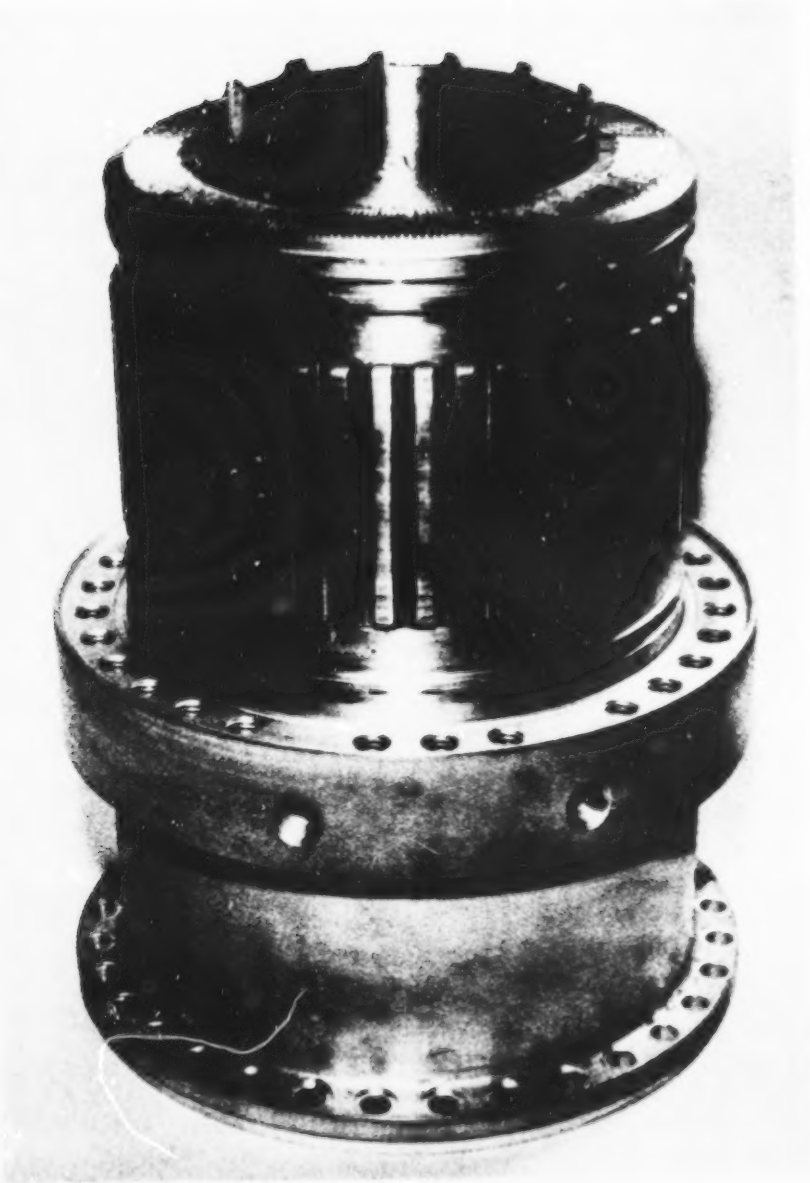

Figure 7 Cooler Sub-Assembly

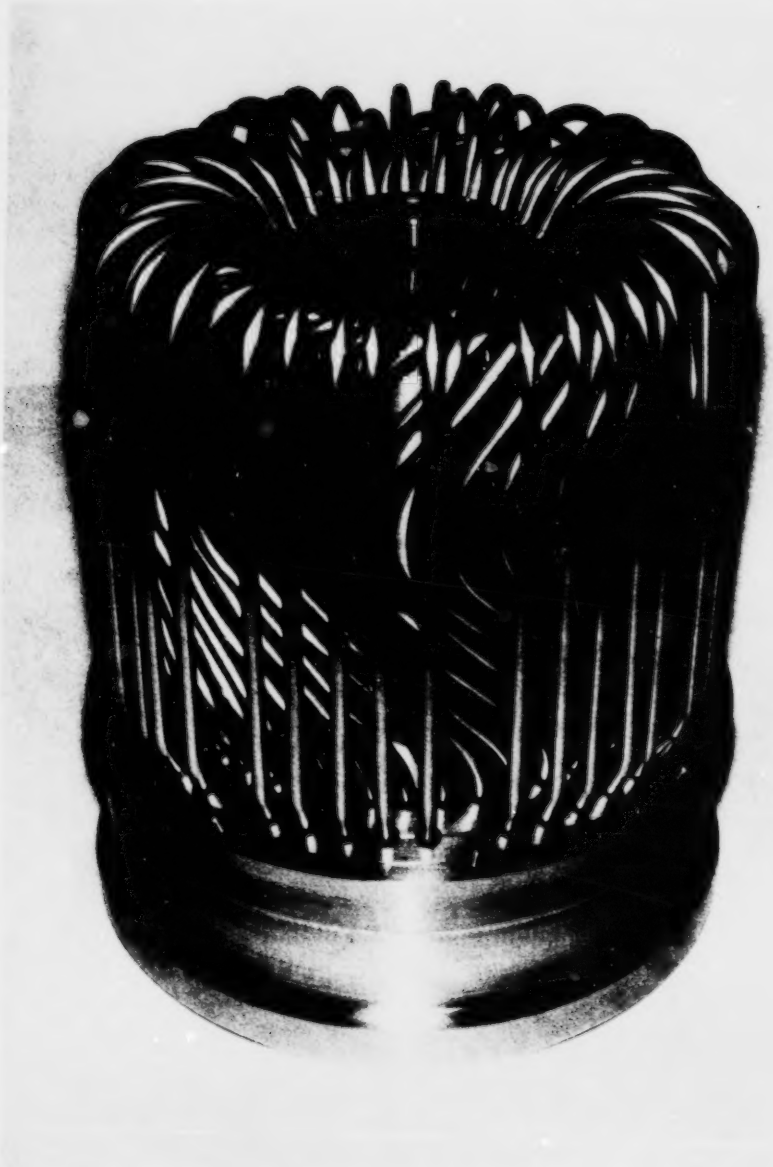

Figure 8 Heater Head

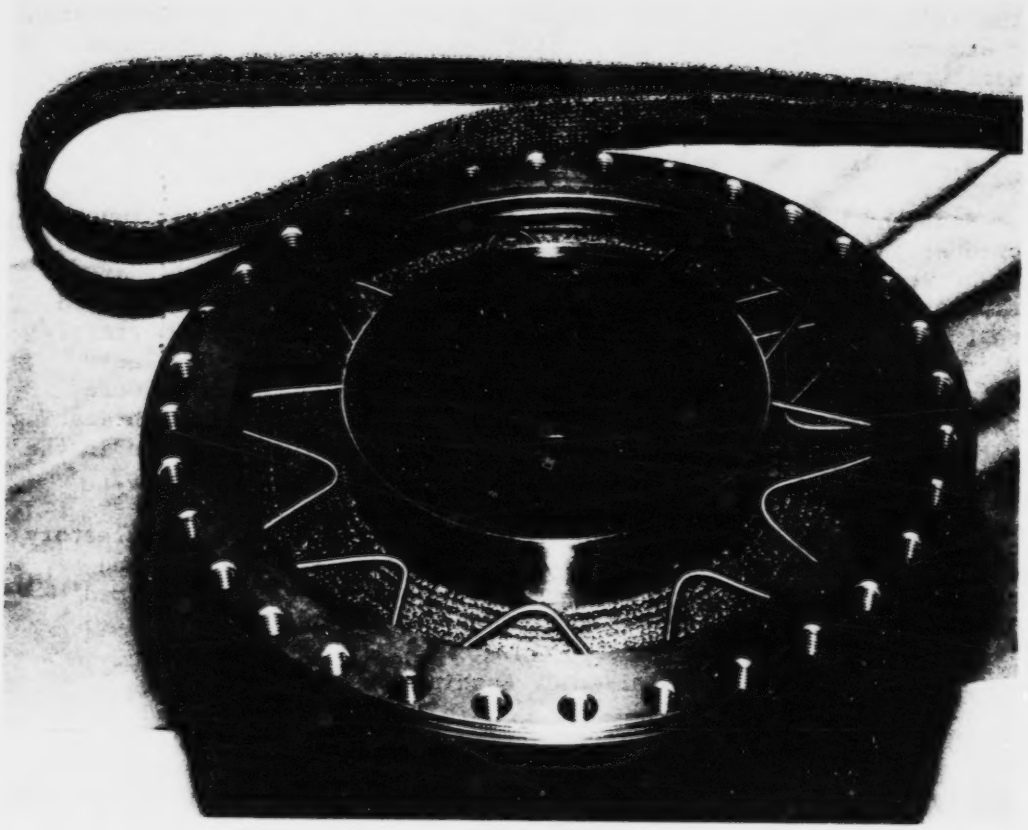

Figure 9 Inverted Heater Dome Showing Regenerator with Retaining Ring and Woven Matrix Material

a) displacer base,
b) displacer rod,
c) gas spring piston,
d) power piston,
e) piston rod.

Only one of the above (the piston rod seal) was required to withstand the high pressure difference between the mean cycle pressure and atmosphere. The others were relatively low pressure-difference dynamic seals some of which also served as guide/wear rings. Differing requirements dictated various configurations described below:

3.2.1 Displacer Seal/Guide Rings

Near the base of the main body of the displacer two rings functioned as guides for the displacer within its cylinder and provided a seal between the hot and cold spaces within the displacer cylinder. The pressure difference across these rings was simply that associated with the pressure drops through the heat exchangers. More importantly the drag associated with these rings was kept low so that displacer motion was not inhibited. The temperatures to which the rings were exposed was variable tending to be rather high. Use of self-lubricating materials was desirable to minimize requirements for lubrication in the displacer cylinder and to avoid contamination of heat exchangers.

Polyphenylene sulphide (PPS) for these seals proved unsatisfactory due to dimensional change with temperature and a consequent marked increase in friction. Graphite, while much more fragile, performed well during engine operation.

3.2.2 Displacer Rod Seal/Guide Bushing

This bushing acted as a low-friction guide for the displacer rod and as a seal between the engine working space and the gas spring assembly, a relatively low pressure difference. Leakage past this bushing was useful and necessary to maintain the mean gas spring pressure equal to the mean pressure in the engine working space.

The mass of the bushing is not significant and simple substantial bushing of graphite proved satisfactory for the engine testing.

3.2.3 Gas Spring Piston Seal

The gas spring piston transmits the gas spring loads to the displacer and must also, intermittently, provide sealing between the spring spaces and the bypass space involving a large number of edge transits. The requirements were met using a Delrin piston with a multi-groove labyrinth seal.

3.2.4 Piston Seal/Guide Rings

The power piston operates in a cool water lubricated environment. The pressure difference is limited to that between the working space and the balance space. There are essentially no side loads on the

piston.

For this application sealing rings of PPS proved satisfactory both when lubricated with water and when operated without lubrication.

3.2.5 Piston Rod Seal

This was a small diameter seal operating in a cool, lubricated environment with very little side force or motion, but was required to sustain the whole pressure difference between the balance space (typically near that of the working space) and the crankcase (i.e. atmospheric pressure). Leakage past this seal represents a direct loss of working fluid from the engine. The seal used was a commercial pressure activated lip seal. The material was a self-lubricating (molybdenum disulphide) impregnated elastomer but was kept immersed in a water bath to enhance lubrication and to ensure that over-heating of the seal against the piston rod did not occur.

3.2.6 Static Seals

'O'-ring seals were used wherever possible for both large or small pressure differences. Additional cost and complexity was involved in their design and manufacture compared with conventional gaskets, but they are more tolerant of misalignment during repeated assembly, and are easily replaced. The pressure seal between the heater assembly and the cooler presented special problems due to the high temperature at the bottom of the regenerator housing where the seal is located. Selection of silicone as the 'O'-ring material along with a ceramic-based back-up material proved a satisfactory seal with no apparent degradation. Neoprene 'O' rings were used in all other locations.

3.3 Bearings and Lubrication

Crankshaft and connecting rod bearings were designed to operate with water lubrication. Materials were selected that had adequate strength and could operate in the water environment with low friction. The actual configuration was constrained to that of the metallic bearings of the Lister diesel engine with discretionary choice of bearing clearances, lubricant distribution grooves, and lubricant feed.

In approaching the design of the bearings the function of each was examined. Bearing duty was described in terms of loading (often time-varying with respect to both magnitude and direction), velocity and the type of lubrication possible. In this context the main crankshaft bearings operated at constant speed and at relatively low bearing pressures. The magnitude and direction variation was sensitive to the extent of pressure balancing. Lubrication could be introduced either through the shaft or through the bearing walls. The latter was preferred for simplicity where load direction was constant.

The gudgeon pin bearing in the crosshead had the highest bearing pressure of all the bearings and the velocity, while low, varied continuously in both directions. Heat generation at the bearing was because of the low velocity, assumed to be insignificant. It was possible

for load reversal to occur depending upon the level of pressure balance.

The big-end connecting rod bearing was exposed to the most severe combination of bearing pressure and velocity. The demands made upon the lubrication of this bearing were the highest and in practice the lubricant could only be introduced through the crankshaft.

The strength requirements for all of the bearings were met by bearings made from 'Ferobestos', asbestos-reinforced thermosetting resin impregnated with colloidal graphite to make the material self-lubricating to some degree.

Bearing clearances were determined in the light of the bearing materials, loads and mode of lubrication. Hydrodynamic lubrication of the main and connecting rod bearings using the existing configuration and bearings loads and with water as the lubricant was infeasible. Even with extremely small diametral clearance (of the order of 0.025 mm) eccentricity ratios were as high as 0.95. In addition to criticality of alignment and finish, maintenance of a constant temperature distribution was essential since changes of dimension with temperature are larger. In a conventionally designed bearing the flow of coolant is a direct function of the clearance and, since a rise in temperature of the bearing will decrease the clearance, the likelihood of thermal instability exists in spite of changes in flow with changes in lubricant viscosity. Hydrodynamic lubrication was not possible and hydrostatic lubrication was precluded due to its complexity so the only possible alternative was boundary lubrication. In this mode the bearings operate with solid-to-solid contact and sliding friction. The principal design considerations for this were wear, friction loading and heating. The materials in the bearing were thus required to resist wear as well as possessing a low coefficient of friction. The principal function of the lubricant was to remove heat from the bearing with some reduction of friction where possible.

To minimize friction as well as to protect the crankshaft from corrosion the bearing surfaces on the crankshaft were plated with chromium and ground to achieve a smooth finish. The bearing sleeves were produced by rough turning the bearings and then soaking them in water until saturated in order to achieve dimensional stability. The bearings were finish turned to a press fit in their housings after which the inner diameters were finished turned. Minimum clearances of 0.002 mm/mm diameter were recommended by the bearing material manufacturer.

The minimum tolerable clearances were determined by constructing bearings with a small clearance and then operating at increasing speeds until seizure occurred. A small increase in the clearance was introduced and running was repeated. With increasing clearance it was noted that the achievable speeds also increased. Seizure of the bearings in every case was due to thermal instability.

Water was introduced and distributed to the bearings by axial grooves slightly short of the edges of the bearings. The best cooling and even lubrication was expected from this arrangement.

The gudgeon pin bearing was designed somewhat differently. Its loading was very high but the velocity was low and so little heat was generated at this bearing. It was therefore possible to use a very small clearance so that load reversals would not result in free movement of the crosshead. To reduce the bearing pressure to a tolerable level the bearing was lengthened by incorporating a steel sleeve in the connecting rod.

Plain water was initially used as the lubricant. It soon became apparent that the steps taken to protect the engine against corrosion were not sufficient and so 0.5% emulsifying oil of the type used in a water-based cutting fluid was added to the lubricating water in what then became a closed-loop lubrication circuit. No change in engine friction was perceived as a result of this changeover; additionally, prevention of the ingress of lubricant to the working spaces had been achieved by that stage of development. Nonetheless, plain water was injected into the high pressure zone to lubricate the piston and piston rod so as to maintain safety and because corrosion protection in this section appeared to be adequate. This resulted in a slight dilution of the lubricant but was negligible over the duration of the test programme.

Lubrication was required in both the crankcase/crosshead region and particularly around the piston rod. The former took place in an environment at atmospheric pressure while the piston and upper portion of the piston rod were operated at the pressure of the working (balance) space. Supply of lubricant to these two spaces was provided by two separate pumps - the low pressure pump being a simple gear-type and the high pressure pump being a piston type. Both were driven by electric motors. Control of flow and pressure within each circuit was achieved using over-pressure protection and manual distribution valves.

3.4 Flywheel Design and Engine Balancing

Construction of the experimental Ringbom-Stirling engine involved addition of the free displacer and its cylinder and a power piston and cylinder as well as the conversion of the Lister engine piston into a crosshead. Balancing of the displacer was deemed both impractical and unnecessary due to the low magnitude of the forces involved. The impracticality arises from the highly non-uniform motion of the displacer. The mass of the power piston added to the higher piston utilized as the crosshead represents a more significant source of vibration and imbalance. In the diesel engine, balance was achieved by the addition of mass to one spoke of each of both flywheels but the addition of further mass to balance the power piston was not permissible because of the increased bearing load rotating at the crankshaft speed.

It was anticipated that initial operation would take place at low speeds, so to increase the flywheel inertia, additional mass was added to the rims of the flywheels in the form of steel rings, engaged the rims of the existing wheels so as not to impose additional loading in a radial direction. The speed of the experimental engine was accurately controlled by the dynamometer, so the necessity for the flywheel mass

might be questioned, but was justified because of the consequent decrease in torque variation. Doubling the inertia of the flywheels was thought to be a reasonable compromise with regard to increased bearing loading and decreased torque variation.

3.5 Firing Equipment

The ultimate objective was to incorporate engine heating by fluidized bed coal combustion with transfer of heat to the engine by a sodium heat pipe system and the engine heater head was designed with this objective in mind. A simpler system of heating by the combustion products of propane and oxygen was adopted for preliminary testing. Reduced effectiveness of the heating system in the preliminary work was accepted as the price for a system that could be quickly and easily implemented.

Thermal flux densities achievable during sodium condensation heat pipe operation are much higher than can be achieved with gas to solid convection heat transfer but large sodium heat pipes were not available and in fact were the topic of a separate but related study (Walker (1983)). Combustion gases were the obvious choice for a simple alternative heat source. High temperature differences and high gas velocity were necessary to achieve the high heat transfer rates with a tubular heat exchanger designed for use with the heat pipe system. With these considerations in mind the heating system adopted involved use of twin oxy-propane heating torches available commercially. The system offered a simple method of control, use of readily available fuel gases and a high temperature of combustion. Control was exercised manually on the basis of direct readings of heater tube temperature. Rapid cooling of the system was possible by shutting off the fuel and opening the dilution air control valve.

4. ENGINE SPECIFICATIONS

Some 106 sub-assemblies, parts and components were separately identified for the prototype engine system. Fabrication and acquisition of these component parts and sub-assemblies continued through 1982 from January to October. Work thereafter until December 1982 was principally concerned with engine fitting and assembly and with installation of the engine test bed and instrumentation. The unit was virtually complete in December 1982 and engine testing commenced shortly thereafter.

Full details of the engine test-bed installation are included on the appropriate drawings on file at the University of Calgary.

This section presents a summary of important engine dimensions and descriptions of key elements as well as design operating conditions.

4.1 Components

4.1.1 Crankcase

- Lister Model 8/1 single cylinder diesel engine crankcase modified for use with water by coating internally with Emralon 333.
- Crankcase was not pressurized.

4.1.2 Crankshaft

- Lister Model 8/1 diesel engine crankshaft modified for use with water by plating journals with chromium followed by grinding to size; remainder of shaft coated with proprietary water-resistant coating.
- Crank throw: 69.85 mm
- Main bearing dia: 50.80 mm
- Crank bearing dia: 63.50 mm
- Main bearing length: 91.4 mm
- Crank bearing length: 59.7 mm

4.1.3 Crankshaft Bearings

- All bearings fabricated from Ferobestos Grade LA3 asbestos-reinforced, colloidal graphite-filled thermosetting resin.
- Main bearings: one-piece bushes, 0.10 mm diametral clearance
- Crank bearing: split bushing, 0.12 mm diametral clearance

4.1.4 Crosshead Bearing

- Fabricated from Ferobestos Grade LA3
- Bearing diameter: 31.75 mm
- Bearing length: 67.31 mm
- Bearing clearance: <0.02 mm

4.1.5 Crosshead

- Fabricated from stainless steel with welded-in gudgeon pin supports.
- Vented construction.
- Contact with cylinder using wear rings of Ryton.
- Cylinder wall coated with Emralon 314 to provide self-lubrication, wear resistance and corrosion protection.

4.1.6 Piston Rod

- Fabricated from stainless steel with threaded connection to crosshead.
- Diameter: 25.4 mm

4.1.7 Piston Rod Seal

- Single lip seal with spring loading operates submerged in water.
- Material: Molythane
  Parker Std. Polypak #S-2500

4.1.8 Piston

- Fabricated from aluminum, attached to rod using tapered collars.
- Diameter/stroke: 114.3 mm/139.7 mm
- Sealing using four Ryton R-5304 split rings against cylinder wall coating of Emralon 333, operable either wet or dry.

4.1.9 Displacer

- Welded assembly comprised of hydroformed dome and vessel, thermal shield, base incorporating sealing ring grooves and stiffening stays, displacer rod, the whole being made of stainless steel; complete assy. includes gas spring piston.
- Diameter/stroke: 127.0 mm/129.7 mm
- Rod diameter: 44.45 mm

- Mass: 1.6 kg

- Pressurization of vessel achieved using diaphragm-type non-return valve.

  Vent diameter: 1.0 mm

4.1.10 Heater

- Upper portion of displacer cylinder incorporates a number of heater tubes which are vacuum brazed into position.

- No. of tubes: 88

- Length of tubes: 300 mm each

- Outside/inside dia: 6.35 mm/4.52 mm

- Material: 18/8, 316 Stainless Steel

4.1.11 Regenerator

- The regenerator is formed by coiling and packing knitted and calendared stainless steel wire into the annular regenerator cavity.

- Annulus outer diameter: 228.6 mm

- Annulus inner diameter: 127.0 mm

- Length: 50 mm

- Wire Diameter: 0.0889 mm

- Fill factor: 20%

4.1.12 Cooler

- Fabricated from machined, formed, and welded aluminum; heat exchanger is contained within a flanged, cylindrical steel pressure vessel; displacer cylinder formed by a stainless steel sleeve pressed into cooler and regenerator units.

- No. of fins: 198

- Groove form: triangular, 25.4 mm x 1.575 mm

- Groove length: 173 mm

- Coolant flow (water) 31 ℓpm

4.1.13 Gas Spring

- Piston and cylinder with centre bypass and variable volume spring

spaces; isolated from displacer cylinder by graphite seal

- Gas spring piston diameter: 63.5 mm
- Free motion of displacer using bypass: 77.5 mm
- Gas spring piston area - upper: 1615.0 mm^2
  - lower: 3167.0 mm^2
- Two stage spring action is in effect: the first stage incorporates a relatively large, adjustable volume, the second stage comes into effect near the end of the stroke and isolates a small fixed air volume.
- Pressure changes within the gas spring associated with displacement by the displacer rod are reduced by the addition of three fixed-volume accumulators to the gas spring bypass space.
- Maximum/minimum spring volume, first stage, upper: 136,600/22,600 mm^3
  lower: 148,800/35,700 mm^3
- Second stage spring volume, upper: 11,300 mm^3
  lower: 22,170 mm^3
- Bypass volume c/w accumulators: 875,000 mm^3

4.2 Operating Conditions

The engine design operating conditions involve setting and control of rotational (crankshaft) speed, heater and cooler temperatures, and mean and maximum pressures in the working space, balance space, and gas spring.

4.2.1 Speed

The engine speed is set and controlled by the dynamometer system and is not to exceed 1000 rpm (16.7 Hz).

4.2.2 Temperature - Heater

Due to degradation of material strength with increasing temperature, heater temperatures (as measured on several tubes and on the vessel) are not to exceed 923 K under any circumstances while at maximum rated pressure.

4.2.3 Temperature - Cooler

Cooling water from the water mains was available at a temperature no higher than 293 K.

4.2.4 Pressure

- Maximum system pressure was limited to 7.24 MPa by rupture discs.
- Design mean working space pressure: 5.00 MPa

5. INSTRUMENTATION

5.1 MACSYM II System

The focus of the data gathering system was a MACSYM II computer made by Analog Devices, Inc. (128 Kbytes RAM) capable of sampling various parameters at differing frequencies, processing the data (computation of average values, computation of temperature using polynomial approximation to thermocouple characteristics, etc.), and generating output displayed on a cathode ray tube or printer.

It also has the capability to generate control signals. This control provision was not utilized, but can be arranged to control the heating of the engine should this be required.

The computer was equipped with the following communication capabilities:

a) Analogue Input used to sample the following:
   (i) Multi-tasking control signals
   (ii) Tachometer signals
   (iii) Working space pressure signals
   (iv) Displacer and piston position signals

b) Analogue Input Card plus Isothermal Termination Panel. These cards provide for fourteen thermocouple input channels each having room temperature compensation and offset capability.

c) Strain Gauge Input Card. The output of the torque-sensing load beam was sensed using this card.

d) Pacer Clock. Allows the sampling of working space pressure and piston and displacer positions to be initiated at piston top dead centre and then sampled at regular angular intervals over a single engine revolution.

Updating of the computer-driven displays was accomplished approximately five times per minute. In addition provision was made for continuous display of instantaneous values of engine speed and torque and the piston and displacer positions.

5.2 Measurement of Engine Operating Parameters

Measurements of engine operating parameters using the MACSYM II system and other means were as follows.

5.2.1 Engine Speed

The test bed for the TDC Ringbom-Stirling engine incorporated a converter/controller and a direct-coupled d.c. motor/generator; the

system incorporated tachometer feedback and consequently achieved speed regulation better than 1%. The feedback signal (400 v.d.c. at 4000 rpm) was used to sample the speed by reducing the voltage potentiometrically. The speed was sampled once per program cycle. Calibration was checked using a B&K strobotachometer.

5.2.2 Engine Torque

The motor/generator on the test bed was mounted on flange-type self-aligning bearings such that any torque reaction during monitoring or generating would be transmitted by a load beam fitted with resistance strain gauges. The full bridge was excited by the same card of the MACSYM II communication card. The beam was calibrated by a dead weight method in conjunction with a direct strain measurement box and the MACSYM II.

5.2.3 Piston Position

The piston position was inferred from the crank position by constructing a small eccentric having the same proportions as the crank/connecting rod of the Lister engine. The position of the eccentric was sensed by a linear variable differential transformer (LVDT) with its own excitation and signal conditioning unit. Top dead centre was sensed using a small optical unit attached to one of the flywheels. This event then initiated a series of thirty six sets of pressure position samples taken at uniform time intervals.

5.2.4 Displacer Position

The position of the displacer relative to the displacer cylinder was sensed directly by an LVDT with the core attached to the displacer rod. The transducer was calibrated by direct measurement using a steel rule. Sampling of the position was synchronised with that of the piston position sampling so as to allow the recording of thirty six samples over one cycle. Continuous display of both piston and displacer positions was accomplished using a dual-beam oscilloscope. Records of these motions were made using a two-channel Hewlett-Packard chart recorder.

5.2.5 Working Space Gas Pressure

The pressures in all three spaces (the working space, the balance space, and the gas spring space) were measured with variable reluctance pressure transducers which passed analogue signals to the MACSYM II. A continuous display of average pressure for each space was also provided using Bourdon tube pressure gauges with a small diameter snubber. The working space pressure was sampled synchronously with the piston position in order to allow reconstruction of a P-V diagram. The sampling program calculated the mean pressure over the engine cycle. There was also provision to record the analogue signal on the chart recorder along with either piston or displacer position. Calibration of the transducers was done using a dead weight calibration device.

5.2.6 Balance Space Gas Pressure

Sensing of the gas pressure in the balance space (i.e. the space beneath the power piston and including the balance space accumulators) was done in the same fashion as the working space. By locating the sensor port at the top of one of the accumulators it would have been possible, given sufficient time, to determine the losses associated with irreversibilities in the balance space. The sample pressures were averaged, however, and a single value displayed on the output. The value displayed, like the working space pressure, was the gauge pressure. A direct reading pressure gauge was also provided.

5.2.7 Gas Spring Space Gas Pressure

In this case, the pressure transducer was arranged to measure the difference between the upper and lower gas spring spaces - virtually zero for most of the displacer stroke because of the piston bypass grooves. A direct reading pressure gauge was connected to the upper spring space. The upper spring space connection was also used for venting the gas spring.

5.2.8 Heater Tube Temperatures

A total of eight thermocouples were attached to the heater tubes. Of these, four were sampled by the MACSYM and four were led to a direct readout system which allows any of these to be displayed by manual selection. The thermocouples are metal-sheathed, ceramic insulated Type K and were installed by stripping the sheathing and insulation from the terminal 5 mm and then welding the 28 gauge wires directly to the tubes in order to minimize the sensing error. The MACSYM output recorded both individual readings of the four heater tube temperatures and the arithmetic mean temperature.

5.2.9 Heater Head Temperature

Two thermocouples of the type described above were attached to the top of the heater head adjacent to the entry points of the heater tubes. These were sampled by the MACSYM and recorded individually.

5.2.10 Regenerator Housing Temperature

Six thermocouples of the type described above were attached to the part of the heater head surrounding the regenerator. Two of these were attached to the large lower flange of the heater head; the remaining four were attached to the outer surface of the regenerator housing wall. Each of these temperatures was recorded individually by the MACSYM.

5.2.11 Coolant Water Flow and Temperature

The cooling water was passed through a rotameter having a capacity of ten gallons per minute. Calibration of the flow meter was by graduated receptacle and stopwatch. The temperature of the coolant leaving the engine was measured using a type K thermocouple recorded

by the MACSYM.

5.2.12 Lubricating Water Supply Pressures

Flow to the rotating bearings in the Lister engine crankcase was manually controlled and monitored using Bourdon tube pressure gauges.

6. PRE-TEST TRIALS

6.1 General

On completion of fabrication and assembly of the components comprising the Ringbom-Stirling engine a number of hours of engine operation were used to acquire experience in the running of the engine, to remedy operational and performance problems, and to gain some early indications of the performance of the engine.

Over the period during which the trials were conducted, changes in several components were implemented including the gas spring assembly, the displacer, the displacer cylinder base, and the power cylinder. These changes allowed the power output from the engine to be increased by gradually raising the heating temperature, mean working space pressure, and RPM.

Operation of the engine system requires diligent monitoring of all systems by an operator who is thoroughly acquainted with the configuration and function of all parts of the engine. Additional manpower is required for the operation of the recording equipment and also for maintenance of a safety surveillance.

Guidelines for operation of the engine system have been established in the form of a basic check list, see Appendix 'A'.

6.2 Operating Data

Over the testing period results were obtained which may be described as either, a) quantitative, or b) qualitative, as outlined below. Progress of the trials may be traced by listing them along with values of key operating variables but preceded by description of the typical output from each test. A summary of the principal results achieved is given in Table 1.

Both the hardware and software were continuously upgraded during the trials so that meaningful comparisons between one test result and another are not possible.

Key parameters are presented in Table 1 for each of the engine tests listed. In addition, the summary indicates the additional records maintained. Chart recordings of piston and displacer positions and working space and/or gas spring pressures were obtained for many tests although only two simultaneous records could be obtained at any time.

The output of the MACSYM computer was developed over the testing period to obtain a suitable compromise between the amount of information presented and the time available for output. Samples of the output in its final form are shown by Figures 10 and 11.

The variables indicated on the computer output are defined as follows:

Variable Name	Explanation
Today	Month Day Year
Time	Hour Minute Second (24-hour clock)
Ref.	Cold thermocouple junction temperature (°C)
T1,T2,T3,T4	Thermocouple temperatures on four heater tubes (°C)
M(Avg.)	Arithmetic mean of T1,T2,T3,T4 (°C)
H1,H2	Thermocouple temperatures at top of heater dome (°C)
F1,F2	Thermocouple temperatures at dome flange (°C)
W1,W3	Thermocouple temperatures at regenerator housing wall - upper (°C)
W2,W4	Thermocouple temperatures at regenerator housing wall - lower (°C)
C1,C2	Thermocouple temperatures on coolant water output (°C)
CSP	Cold (working) space pressure - average of 36 values (psi)
BSP	Balance space pressure - average of 36 samples (psi)
CSPP	Peak cold space pressure (psi)
GSPP	Peak gas spring pressure (psi)

Piston position and displacer position were sampled 36 times per crank revolution with the first sample occurring at crank top dead centre. The values may be converted into distances but calibration was done such that equal changes in position units reflect equal values of swept volume.

6.2.1 Chart Output

Analogue output in the form of chart recordings of piston and displacer motion is illustrated in Figure 12 below which shows results from three different tests. The displacer motion was influenced by engine speed, mean working space pressure, and gas spring pressures. The three tests shown were recorded at increasing levels of speed in the range 160-410 RPM.

6.3 Analysis of Operating Data

To achieve displacer motion approaching that of the prediction models it is necessary to ensure the pressure variation of the gas spring associated with the displaced volume of the displacer rod is reduced to a tolerable level.

Design and construction of the displacer is critical. The hollow vessel must be pressured to prevent collapse and free motion is required to achieve proper running. The displacer mass must be minimized in order to improve the engine power by allowing the engines to run at reasonably high speeds.

In the experimental engine friction in the running gear was very high. This was determined by measuring the shaft output power for a particular trial and comparing it with the indicated power determined from an engine P-V diagram. For Run 2 (28 March 1983) the brake power was 1.24 kW compared with an indicated power of 2.0 kW. Separate tests

performed by motoring only the crankshaft also indicated values of friction coefficient much higher than expected.

It was found that during operation of the engine adjustment of the gas spring pressure was particularly important to achieve the desired displacer motion even when the engine speed and mean working space pressure were well matched.

The Beale equation indicates in the most general terms that the power output of a Stirling engine increases with operating pressure, speed, and temperature difference. If the operating temperatures for a particular engine were to remain constant, the power output should vary linearly with speed and linearly with pressure.

The Ringbom-Stirling engine is a special case since the phase angle between the piston and the displacer (or some equivalent factor since the displacer motion is typically not the same form as the piston motion) depends upon the relationship between speed and pressure. Parametric studies using the digital simulation program developed for this project indicated that, for a given pressure, the maximum engine power occurred at a specific speed.

With this in mind Figure 13, shown overleaf, was plotted to indicate the distribution of results obtained from tests which produced more than 0.10 kW. Had all tests been performed at the same temperature conditions it is expected that a gradual increase in power output as pressure and speed followed would have been obtained. The figure shows that this was not the case and rationalization of the results on the basis of correction for temperature variation does not improve the consistency of the results. It was concluded that a number of unknown variables influencing the engine performance were not under experimental control during the tests.

Future measures could include a more detailed and systematic examination of the operation of the engine. Areas of interest are:

a) Mechanical - the interaction of self-lubricating solids and water lubricant is not well understood. More direct experiments will reveal means by which design of this type of bearing may be substantially improved.

b) Fluid dynamics - the flow patterns within the engine are complicated. Problems to be addressed include examination of gas spring and balance space irreversibilities and means by which better flow passages may be designed and manufactured.

c) Optimum heat exchanger design remains a continuing focus of attention since designs are also subject to pressurization loadings and difficulties of manufacture as well as reduction of dead volumes.

d) Displacer motion control is clearly of prime importance. In the Ringbom-Stirling engine the basic problem is complicated by the variable speed capacity of the engine.

e) Engine control systems - the sensitivity of engine performance to the operating conditions and the dangers involved in straying from the design conditions both represent major areas of concern.

RESULT SUMMARY

Date	Run No.	Seq. No.	RPM	CSP	BSP	M	Torque[1]	kW[1]	Position Record Piston	Position Record Displacer	Pressure Record Wk'g. Sp.
20 Jan.	1	1	74	0	0	16	0.0	0.00	x	-	x
	2	2	113	0	0	15	-1.3	-0.02	x	-	x
	3	3	178	0	0	15	-1.4	-0.03	x	-	x
	4	4	301	0	0	16	-0.7	-0.03	x	-	x
	5	5	492	0	-1	16	0.0	0.00	x	-	x
	6	6	104	31	5	17	-7.3	-0.11	x	x	x
	7	7	188	33	6	17	-5.1	-0.13	x	x	x
	8	8	297	34	6	18	-4.4	-0.19	x	x	x
	9	9	494	35	7	18	-4.4	-0.31	x	-	x
	10	10	123	83	4	14	-46.1	-0.81	x	x	x
	11	11	182	83	5	12	-30.7	-0.79	x	x	x
	12	12	309	83	5	8	-24.9	-1.09	x	x	x
	13	13	492	86	5	1	-24.2	-1.69	x	x	x
	14	14	478	94	2	-17	-32.2	-2.19	x	x	-
21 Jan.	1	15	39	1	-1	17	-6.6	-0.04	x	-	x
	2	16	-	-	-	-	-	-	x	x	x
	3	17	174	35	29	19	-2.2	-0.05	x	x	x
	4	18	301	36	32	20	-1.5	-0.06	x	x	x
	5	19	490	39	38	20	-3.7	-0.25	x	x	x
	6	20	113	65	65	17	-12.4	-0.20	x	x	x
	7	21	178	64	67	16	-8.1	-0.20	x	x	x

es [1]) Negative values of torque and power indicate that the engine is being motored by the dynamometer.

Table 1: Summary of Experimental Results, Part I

RESULT SUMMARY (continued)

Date	Run No.	Seq. No.	RPM	CSP	BSP	M	Torque[1]	kW[1]	Position Record Piston	Position Record Displacer	Pressure Record Wk'g. Sp.
	8	22	293	64	69	13	-8.1	-.33	x	x	x
	9	23	492	67	74	8	-13.9	-.97	x	x	x
	10	24	156	247	256	8	-44.6	-.99	x	x	x
	11	25	320	253	281	5	-43.9	-3.00	x	x	x
	12	26	500	266	280	-3	-54.9	-3.90	x	x	x
2 Feb.	1	27	166	1	-1	16	0.7	.02	x	x	-
	2	28	254	1	-2	15	0.7	.02	x	x	-
	3	29	256	38	43	13	-5.1	-.19	x	x	-
	4	30	254	74	86	4	-13.0	-.48	x	x	-
	5	31	170	82	88	3	-15.0	-.37	x	x	-
	6	32	164	73	72	9	-11.0	-.26	x	x	-
	7	33	164	76	75	604	5.5	.13	x	x	-
4 Feb.	1	34	168	51	48	10	-8.5	.01	x	x	x
	2	35	162	55	48	534	1.5	.00	x	x	x
	3	36	168	55	48	608	3.8	.00	x	x	x
2 Mar.	1	37	160	124	129	6	-23	.00	-	-	-
	2	38	160	144	151	13	-28	-.01	-	-	-
	3	39	170	154	151	599	1.08	.00	-	-	-
28 Mar.	1	40	326	212	199	554	8.87	0.41	-	-	-
	2	41	438	360	340	588	20.0	1.24	x	x	-
29 Mar.	1	42	352	391	378	455	2.7	0.14	x	x	-
	2	43	410	326	313	568	12.4	0.72	x	x	-
	3	44	479	334	307	607	13.9	0.95	x	x	-
	4	45	514	390	336	605	10.0	0.73	x	x	-

Table 1: Summary of Experimental Results, Part II

RESULT SUMMARY (continued)

Date	Run No.	Seq. No.	RPM	CSP	BSP	M	Torque[1]	kW[1]	Position Record Piston	Position Record Displacer	Pressure Record Wk'g. Sp.
30 Mar.	1	46	480	367	336	-9	-7.4	-5.06	x	x	-
	2	47	516	181	181	545	3.4	0.25	x	x	-
	3	48	514	374	328	539	9.0	0.66	x	x	-
4 Apr.	1	49	430	217	186	-6	-37.5	-2.29	x	x	-
	2	50	365	271	264	571	18.8	0.97	x	x	-
	3	51	438	266	243	603	15.3	0.95	x	x	-
10 May	1	52	281	247	233	-7	-40.4	-1.61	x	x	-
	2	53	353	271	261	506	18.9	0.95	x	x	-
	3	54	471	345	315	522	16.5	1.10	x	x	-
	4	55	395	417	388	548	23.3	1.31	x	x	-
	5	56	553	490	396	491	2.1	0.17	x	x	-
	6	57	395	400	354	521	13.4	0.75	x	x	-
16 May	2	58	418	378	350	529	18.4	1.09	-	-	-
	1	59	412	353	313	547	16.0	0.94	x	x	x
17 May	1	60	410	332	309	610	23.9	1.39	-	-	-
	2	61	424	371	344	660	30.8	1.85	x	x	x

Table 1: Summary of Experimental Results, Part III

```
STIRLING CYCLE ENGINE TEST DATA
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PROGRAM TAPE 16 FILE 6                    RUN 1.

TODAY-  5 17 83 TIME-   13 45 31

REF.=    20.7856 HTR.TUBES-T1=    592.043 T2=      516.462
T3=      675.84 T4=       657.023 M(AVG)= 610.342
HTR.HD.-H1=       494.091 H2=      565.757
HTR.FL.-F1=       584.85 F2=       404.875
REG.WALL-W1=      344.074 W2=      134.416 W3=      327.798
W4=      85.8649 Z(AVG)= 223.038
CO.WTR.-C1=       12.1474 C2=      10.28627

RPM=     410.156 GEN.TORQUE=      23.8836 FT.LB.KW=        1.3914

C.S.P.(P1)AVG=   332.533
B.S.P.(P2)AVG=   309.41 G.S.P.(P3)AVG=    12.9147
C.S.P.RPM=       421.875
1        414.063
2        426.27
3        428.711
4        423.34
5        408.203
6        391.602
7        375.977
8        362.793
9        350.586
10       340.82
11       330.078
12       321.289
13       312.5
14       303.711
15       296.387
16       289.063
17       283.203
18       277.832
19       272.949
20       269.531
21       266.602
22       266.113
23       269.043
24       273.926
25       279.785
26       286.621
27       294.434
28       303.223
29       312.988
30       323.73
31       335.938
32       348.145
33       361.328
34       374.512
35       387.207
36       400.879
```

Figure 10 Sample of MACSYM Data Output, Part I

C.S.F.P.(P4)=	428.711	B.S.P.P.(P5)=	380.371	G.S.P.P.(P6)=	380.371
CRANK POSITIONRPM=	419.922			DISPL. POSITIONRPM=	419.922
1	53.5719			1	-4.02832
2	52.8493			2	11.3525
3	50.2216			3	24.9023
4	45.7546			4	26.001
5	39.6452			5	17.1814
6	32.0578			6	10.5591
7	23.2879			7	8.66699
8	14.1895			8	9.49097
9	4.82837			9	11.7493
10	-4.72983			10	14.6179
11	-14.1238			11	16.4795
12	-22.8937			12	15.686
13	-30.7439			13	13.092
14	-37.7073			14	9.39941
15	-43.3568			15	4.18091
16	-47.7253			16	-2.62451
17	-50.78			17	-10.6506
18	-52.7836			18	-20.1111
19	-53.4734			19	-30.5786
20	-52.8493			20	-41.8396
21	-50.9114			21	-52.8259
22	-47.791			22	-59.021
23	-43.6524			23	-59.1431
24	-38.1014			24	-57.8308
25	-31.4665			25	-57.1289
26	-23.8463			26	-57.312
27	-15.3391			27	-57.5256
28	-6.01083			28	-57.4646
29	3.41599			29	-56.7932
30	12.9085			30	-55.0842
31	22.2039			31	-52.2766
32	31.0067			32	-48.7366
33	38.7255			33	-44.3115
34	45.032			34	-37.4756
35	49.6633			35	-27.771
36	52.5537			36	-15.625

Figure 11 Sample of MACSYM Data Output, Part II

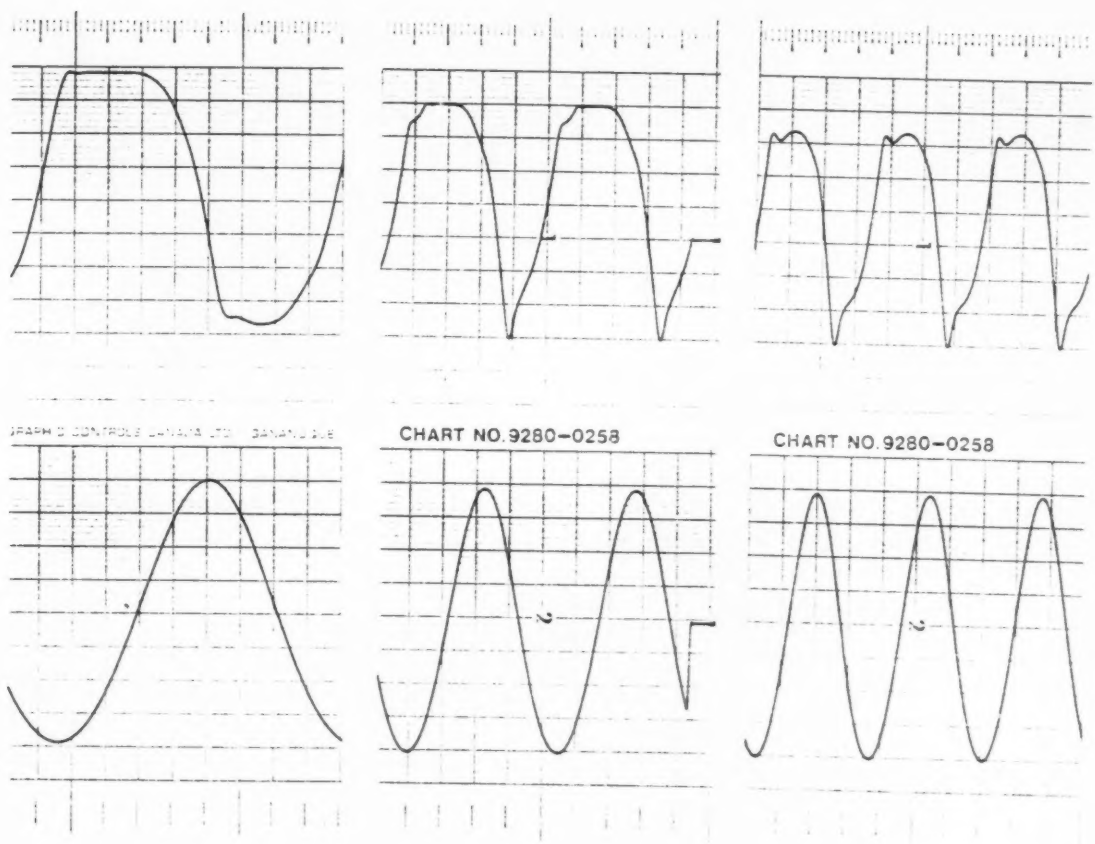


Figure 12 Recorded Motion of Piston and Displacer for Three Different Tests. Displacer motion (upper) and piston motion (lower) are in the same senses as the traces with TDC being uppermost.

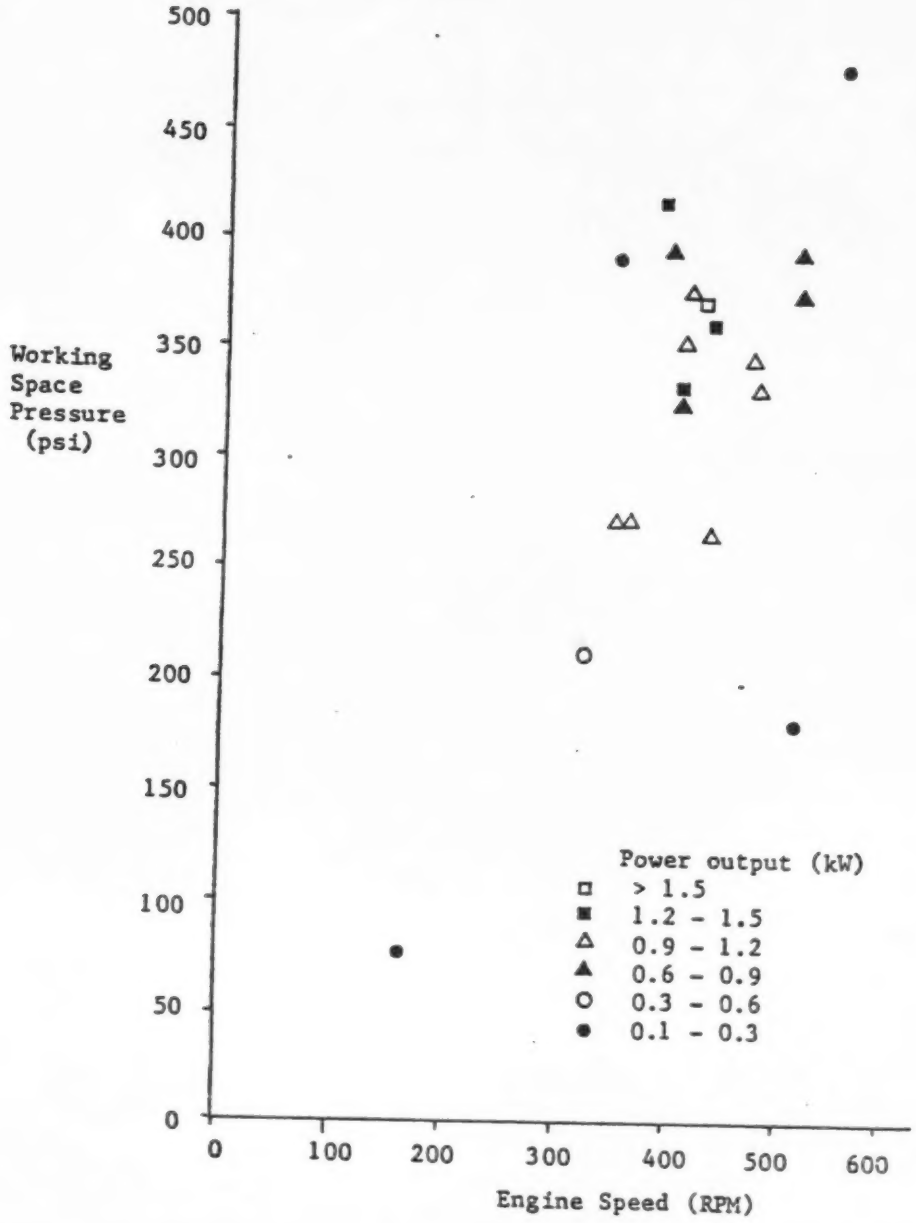


Figure 13 Measured Power Output as a Function of Engine Speed and Mean Pressure

7. COMMENTARY AND CONCLUSIONS

This project is one manifestation of the renewed interest in 'alternative energy' sources brought about by gross increases in oil prices in the early 1970's. Coal was recognised as a fuel that is widely abundant throughout the world in politically stable areas. During the decade following the 'energy crisis' substantial effort was invested in systems using coal both directly and indirectly. Direct use of coal in Rankine and Stirling engines was attractive to minimize processing costs. The price to be paid however was the inconvenience of handling coal compared with oil.

It was thought likely that future coal-fired Stirling engines would probably be limited to large units where the difference in the price of coal, and its local availability would justify the inconvenience of its use or the costs of handling equipment. The possibilities for use included stationary power, railway locomotives, marine engines, off-highway vehicles for mining, forestry, construction and agriculture in a power range 0.5 to 5 MW.

Support for this definition of the applications of Stirling engines came with the appreciation that large diesel engines were slow running with low power ratings in terms of output per unit piston displacement. Whereas automotive diesel engines have a power density in the range 35 to 50 kW/litre (50-70 brake horsepower per litre) engine displacement, larger engines (greater than 1000 kW) (1500 horsepower) have power densities very much less, in the range 15 kW/liter (20 brake horsepower per litre) piston displacement and operate at low speeds, 16 Hz (1000 revolutions per min). Large marine diesels, direct-coupled to the propeller shaft and operating at 1.6 Hz (100 revolutions per minute) have power densities of 1.5 kW/liter (2.0 brake horsepower per litre) piston displacement.

From earlier work on Stirling engines it was known that power densities and engine speeds comparable to the larger diesel engines could be achieved with air as the working fluid.

In the 1950's there was a move away from air to the light gases, helium and hydrogen. This permitted elevation of the speeds and power densities achievable with Stirling engines to the levels achieved with diesel engines in automotive application. This move raised a seal problem that has not been solved despite 30 years of continuous research and development effort by highly competent teams of engineers and scientists. It is simply not possible with existing technology to contain the light gases at high pressures with dynamic seals operating at high frequency with low power losses and with a long operating life.

For large coal-fired Stirling engines a return to the use of air as the working fluid was indicated particularly since the use of light gases offered no advantage at the low speeds and power densities required. The use of air was attractive because it is readily available

all over the world and any leakage from the engine could be readily replenished by a small engine driven compressor. Air is easier to seal than hydrogen or helium and there is no need to have a large compressed gas storage bottle for periodic replenishment.

The problem with air as the working fluid is that leakage of lubricating oil into the working space could result in an explosive mixture of oil and compressed air in the heater tubes or other hot parts of the engine. The resultant explosion, known as 'dieseling' would destroy the engine. A solution to this problem appeared to be the use of water as the lubricant. Lubrication conditions in Stirling engines are much less severe than in diesel engines. The rates of pressure change are orders of magnitude less and no 'top ring' lubrication problem exists in the Stirling engine. The presence of water in the working space would not affect the engine operation. In fact it could be positively beneficial to lubricate seals and piston rings, converting the seals to hydraulic rather than pneumatic further decreasing the already reduced leakage of air compared to light gases. Research on the use of two-phase, two component (air-water) compound working fluids indicated the future possibility of water injection to the engine hot space as a way to boost the power density during vehicle acceleration or other periods requiring maximum vehicle acceleration.

Given the prospect of a shortage of oil it appeared worthwhile to explore the conversion of diesel engines to operate as Stirling engines. There are several ways that diesel engines could be converted for the elements of a Stirling engine can be arranged in a surprisingly wide variety of ways. Some of the variations were explored by Walker (1981a) with particular reference to the MLW Series 251-C 16 cylinder 3000 h.p. Vee diesel engine. Most of the schemes investigated involved the use of two engine cylinders to achieve a single Stirling system, one acting as the power piston cylinder, the other as the hot displacer cylinder. Thus to achieve the same output from the engine it would be necessary to double the power density from the eight power piston cylinders. This did not appear to be feasible. The attractive alternative was to install additional displacer cylinders each coupled to several of the existing 'power piston' cylinders having the appropriate phasing. Superficially it appeared possible to convert the engine and to reinstall it in the locomotive with the additional equipment required and achieve the same engine output as in the original form. To avoid the complication of adding a mechanical drive system for the displacers the principle was adopted of the free displacer/crank controlled piston espoused in the Ossian Ringbom patent of 1907. Small model engines of this type worked well and encouraged the possibility that larger engines of the type could be made.

All the early work on this concept assumed the use of the diesel engine piston as the power piston of the Ringbom-Stirling. This required piston seals capable of absorbing the entire pressure difference of the working space to crankcase (atmospheric) pressure. It was simply not feasible to pressurize the existing diesel engine crankcase to any significant degree.

Analysis of this arrangement quickly showed very high loads on the

running gear would give rise to unacceptable friction losses and that satisfactory lubrication with water was probably not possible.

To alleviate this problem a power piston/piston rod could be added to the diesel piston with the space below the power piston sealed and pressurized to the minimum cycle pressure, the concept of the 'pressure-balanced piston'. This is the arrangement shown in Figure 6. The concept could be extended further by using the space below the piston as the compression space of a second Stirling engine system operating 180° out of phase to the system above the piston. Moreover with a separate power piston it would be possible to increase the diameter of the new power piston to more nearly approach the large bore/small stroke characteristic favoured for Stirling engines to minimize the thermal losses.

Slow speed diesel engines characteristically have a small bore/long stroke characteristic to facilitate fuel spray distribution and combustion. An example is the Lister engine used for the work reported here, the bore was 114.3 mm (4.5 ins), the stroke was 133.25 mm (5.25 ins). This is very different to the bore/stroke ratio of about 3:1 believed to be near-optimum for Stirling engines. The addition of a separate power piston to the long stroke piston of a diesel engine inevitably results in a long thin or tall engine such as that shown in Figure 6. It would of course be possible to arrange the displacer cylinder horizontal but this introduced gravitational complications in displacer operation. One of the basic percepts of free piston engine design is to strive for vertical cylinder arrangements where possible (Walker and Senft (1985)).

One of the unfortunate consequences of adding a power piston to the reference MLW series 251-C diesel engine was that it would almost certainly not fit back into the locomotive engine compartment.

Nevertheless a decision was made to proceed with the proof-of-principle demonstration engine. Moreover, it was decided, arbitrarily, to make the diameter of the added power piston the same as the diesel piston passing up the opportunity for an increase in diameter. In multiple cylinder engines the increase would be limited by the cylinder centreline distances. In a single-cylinder engine the maximum cylinder diameter is limited by stress, balance and lubrication considerations. The effective piston displacement increases as the square of the diameter whereas the hoop stress and the axial thermal conduction loss increase only linearly with the diameter. Thus it is advantageous to increase the cylinder diameter as far as possible to the optimum bore/stroke ratio of about 3:1.

The lengthy period of design, manufacture and assembly of the prototype engine was occupied also with intense intellectual effort on the topic of Ringbom-Stirling engines, the Lister diesel conversion in particular and more generally on the whole field of large coal-burning Stirling engines.

One important product of this intellectual effort was the development of the Senft Theory for Ringbom-Stirling engine operation, (Senft

(1981)). For the first time it was possible with this theory to predict the motion of the free-displacer and, very importantly, to define the limiting frequency of the 'over-driven' mode of operation during which the engine converted from, onomatopoeically, a 'bang-bang' engine to one with harmonic displacer motion, thought to be the maximum operating frequency of the Ringbom-Stirling engine.

A digital simulation program for Ringbom-Stirling engines was developed shortly thereafter (Srinivasan (1982)) and provided the possibility for parametric exploration of the significant engine design parameters. Much further work remains to be done in this field but an important result obtained was an appreciation of the importance of the mass of the displacer in the dynamics of Ringbom-Stirling engines. Redesign of the prototype engine displacer in an effort to reduce the mass so as to achieve an arbitrary power density target of 17 brake horsepower per litre displacement resulted in a structurally inadequate element.

A notable consequence of this particular aspect of the programme was the conception of novel and superior displacer configurations (Walker *et al* (1983), Fauvel *et al* (1984)).

Practical difficulties were encountered in starting and controlling the engine. With the small prototype engine coupled to an electrodynamometer starting was relatively easy using the dynamometer to provide power. However the problems encountered focussed attention on the question of starting large engines already pressurized to operating levels. Speed control achieved by varying the stiffness of the gas springs appeared with test-bed experience much less attractive than at the design stage. Therefore an early concept due to William Martini (Walker (1980)) for an independently-driven displacer was combined with the concept for a Ringbom free displacer to a new form of engine, a Martini-Ringbom-Stirling or Fauvel-Stirling. In this arrangement the displacer is independently driven by a small, variable speed, direct current motor. Oscillation of the displacer causes pressure perturbations in the system acting on the piston, moving it in the cylinder and so starting the engine. The power necessary to drive the displacer is quite small since there is no pressure difference across the two faces apart from the pressure drop in the heat exchangers.

Engine speed can be increased by driving the displacer faster and, of course, the displacer driving power requirement increases with speed as a function of the square of the speed. The relatively high displacer driving power can be provided to an increasing degree as the speed increases by a Ringbom-style spring piston and large diameter displacer rod. Pressurizing the spring space causes the displacer to be driven more by Ringbom free-displacer action than by Martini driven displacer action. Energy to drive the displacer comes of course from the heat input to the engine at high temperature.

A simple power equation for Stirling engines devised by William Beale (Walker (1980)) is:

$$P = 0.015 \bar{p} V f$$

where

P = power output (watts)
$\bar{p}$ = mean cycle pressure (bar)
V = power piston displacement (cm^3)
f = frequency (Hz).

The power piston displacement, V, is clearly an important constituent of the power equation. In traditional designs of Stirling engines it is surprising how little of the actual engine envelope is given over to piston displacement, far less than is devoted to the kinematic drive mechanism, the running gear, or the hollow displacer dome and the combustion space.

In a new departure E.R. Bingham devised a form of engine with very large pistons. The engine crankshaft passes through the centre of pistons along the transverse axis and is attached to the pistons by a modification of the well known Scotch-yoke mechanism. The piston displacement of this type of engine is now very large so that the pressure level or the engine speed can be correspondingly reduced in accordance with the Beale power equation given above.

One conceptual design that combines Bingham-pistons, a hollow Kentfield displacer and a Fauvel displacer drive is shown in Figure 14. This engine was described by Walker *et al* (1984) and is under active investigation at the present time. Another form of engine incorporating Bingham pistons is shown in Figure 15. This is a four cylinder engine of the Siemens-Stirling arrangement in which the cold space of one cylinder (at the top in Figure 15) is coupled through a cooler, regenerator and heater to the hot space of an adjacent cylinder. In this way the number of reciprocating elements per Stirling engine system can be reduced from two to one. The number of cylinders can range from 3 to 6. This type of engine is also the subject of active investigation at the present time.

Little or no work has been done on the coal combustor and thermal transport system for large coal-fired Stirling engines.

One attractive possibility is to combine a fluidized bed coal combustor with a sodium heat pipe thermal transport system. The fluidized bed coal combustor with a limestone feed can use 'dirty' coal with a high sulphur content with little or no harmful emissions. It operates at a temperature of 800°C or thereabouts and extraordinarily high rates of heat transfer can be achieved.

Use of a sodium heat pipe to transport thermal energy from the fluidized bed to the Stirling engine is advantageous from a number of aspects. First it permits great flexibility in the geometry of the fluidized bed/engine arrangement. Sodium heat pipes up to 5 m in length have been demonstrated with only minor heat losses and decrease in temperature along the length. Use of a heat pipe permits transmission flux densitities that are orders of magnitude greater than can be achieved by simple thermal conduction using the purest copper or silver conductors.

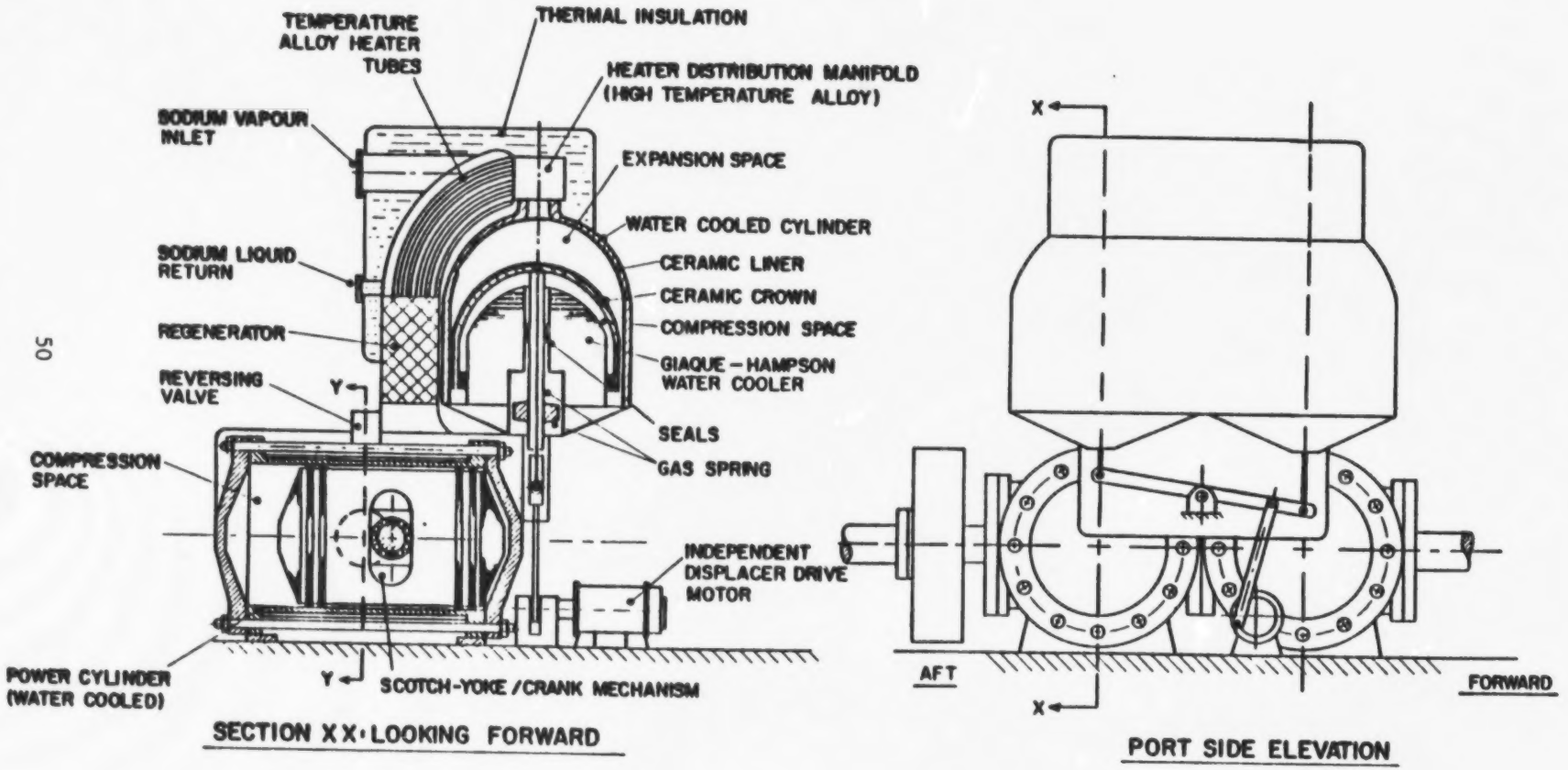
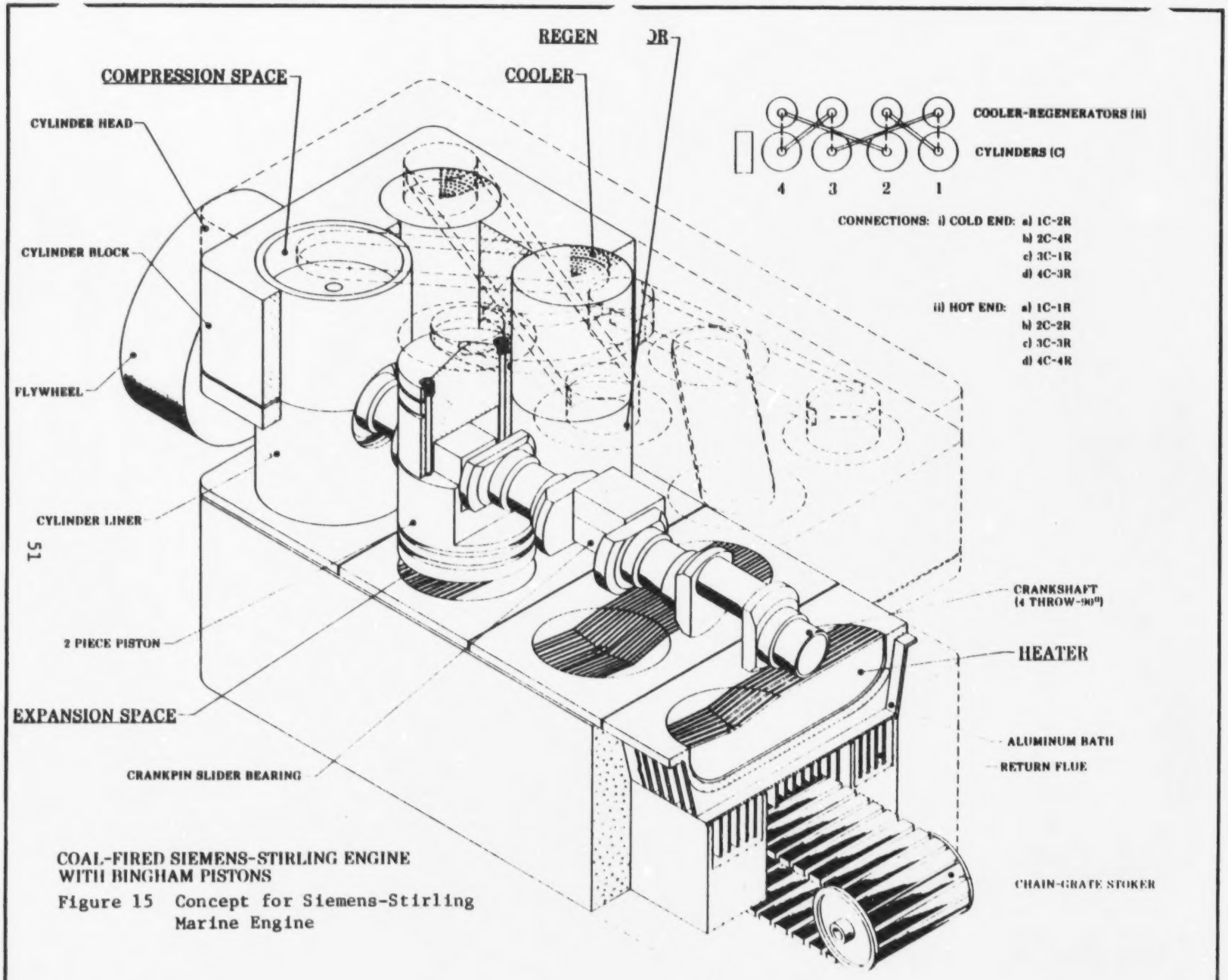


Figure 14 Concept for Bingham-Stirling Marine Engine



COAL-FIRED SIEMENS-STIRLING ENGINE
WITH BINGHAM PISTONS

Figure 15 Concept for Siemens-Stirling Marine Engine

The thermal fluxes achieved in evaporation and condensation of the sodium are very high. The high condensing heat flux permits the heater tubes of the engine to be reduced in diameter and length. This reduces the internal void volume, the dead space of the engine and in consequence increases the power output of the engine. There are no hot spots on the heater tubes because the temperature of the condensing sodium is the same all over the head. Thus the *mean* temperature of the heater tubes become also the *maximum* temperature and may be increased to a prudent fraction of the metallurgical limit with beneficial effect on both the efficiency and power output of the engine.

A two stage heat pipe would likely be used in practice. Multiple first stage heat pipes would convey heat from the bed to an intermediate interface where condensing sodium in the first stages would evaporate sodium in a single second stage system. Separate first stage heat pipes would permit punctures or leaks of sodium caused by the aggressive, hostile environment of the fluidized bed to be tolerated while maintaining operations albeit at a reduced level.

A gas ballasted heat pipe for the second stage would provide the mechanism for automatic control of the temperature level and a safety 'thermal dump' in the event of unscheduled engine shutdown or seizure. Various aspects of sodium heat pipe application to Stirling engines including the design of a test unit for the prototype proof-of-principle engine design were given by Walker (1983).

The engine shown in Figure 14 can be arranged to receive heat from a condensing sodium heat pipe, probably used in conjunction with a fluidized bed combustor.

The alternative method of coal-firing a Stirling engine by a chain grate stoker and direct radiant and convective heat transfer is illustrated on the engine shown in Figure 15. A molten aluminum bath is provided here to act as a thermal flux transformer accepting a low flux/high area input to the bath and providing a high flux, minimum area constant temperature feed to the engine heater.

The recent successful development and operation of six coal-fired ore-carrying ships for Australian coastal service with unmanned machinery compartment encourages the belief that, at least for large engines, there are little or no technical impediments to the direct use of coal as a fuel.

Economic adjustments and improved conservation measures in the decade following the energy crisis have eased the urgency of the situation to the extent that there is presently an 'oil glut'. This is expected to be a temporary phenomenon. In the longer term the future use of coal as a major fuel remains highly probable and it is certain that large coal-fired Stirling engines will find increasing application as power systems in a variety of applications.

China is well endowed with coal resources and to a lesser extent with oil. Strenuous efforts are being made by the Government of China to carry out the industrialization of China on a coal-burning rather

than an oil-burning base. To this end several major research/development engineering groups have been established at the Shanghai Marine Diesel Engine Research Institute, the Institute for Engineering Thermophysics, Beijing and the Institute for Water Transportation, Wuhan. The development of large coal-fired Stirling engines is expected to result from these substantial Chinese efforts.

Major programmes on Stirling engine developments are in progress elsewhere. Many groups and organizations are actively concerned with Stirling engine developments in Japan. Other long-term activities are in progress in Sweden, Germany, Holland, Gt. Britain, Italy and France. Major work is being done in the USSR. Minor programmes are in progress in the United States.

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APPENDIX A - BASIC CHECKLIST FOR ENGINE START-UP AND SHUT-DOWN

Preparation for Start-Up

1. Electricity Supply	
a) control station supply plug (west wall)	IN
b) lubrication pump (L/P) plug (220v)	IN
c) computer plug	IN
d) computer battery charger switch	ON
e) dynamometer, blower, H/P pump breakers	CHECKED
f) igniters	CHECKED
g) solenoid drain valves	CHECKED
2. Water Supply	
a) cylinder drain valves (bypass)	OPEN
b) supply valve (east wall)	OPEN
c) supply valve (control station)	OPEN
d) main cooling supply valve - Rotameter reading	100%
e) lubricant coolant flow	CHECKED
f) H/P pump pressure control valve	OPEN
3. Instrumentation	
a) computer power	ON
b) sampling program	LOADED
c) multi-task power	ON
d) crank position sensor power	ON
e) digital temperature readout	CHECKED
f) LVDT power supplies	ON
g) pressure transducer power supply	ON
h) oscilloscope power	ON
i) oscillograph power	ON
j) oscillograph paper quantity	CHECKED
4. Nitrogen Supply	
a) supply system vent valve	OPEN/CLOSE
b) engine vent valves (working space, bal. space, G/S)	OPEN
c) engine supply valves	CLOSED
d) system supply valve	CLOSED
e) supply system pressure regulator	ZERO PRESS.
f) bottle pressures	CHECKED
g) supply bottle valve	OPEN
h) pressure regulator	SET(≈100 psi)
i) system supply valve	OPEN
j) system supply pressure gauge	CHECKED
5. Security	
a) minimum two persons present	CHECKED
b) fire extinguishers present	CHECKED

6. Fuel Supply	
a) torch valves	CLOSED
b) propane regulator, bottle connections	CHECKED
c) oxygen regulator, bottle connections	CHECKED
d) propane regulator	SET (10 psi)
e) oxygen regulator	SET (43 psi)
7. Gas Spring	
a) spring space pistons	SET
b) positioning valves	CLOSED
8. Instrumentation Checks	
a) position transducers	CHECKED
b) computer monitor mode	CHECKED
c) computer print mode	CHECKED
d) torque transducer (motor/generate)	CHECKED

Start Sequence

1. Dynamometer	
a) dynamometer gain switch (inside cabinet)	'LOAD'
b) speed control	SET ('0')
c) dynamometer power	ON
2. Lubrication System	
a) L/P pump	ON
b) main bearing supply pressure	CHECKED (>75 psi)
c) connecting rod bearing supply pressure	CHECKED (>75 psi)
d) H/P pump	ON
e) H/P supply pressure (at control station and bearing pressure panel)	CHECKED (<100 psi)
f) cylinder water level control	CHECKED
3. Rotation	
a) engine speed control	SET
b) 'MOTOR' light	CHECKED
4. Engine Pressurization	
a) cylinder drain valves (bypass)	CLOSED
b) engine vent valves	CLOSED
c) working space supply valve	OPEN
d) gas spring supply valve	CRACKED (to maintain displacer motion)
e) pressure regulator valve (N_2)	SET (to minimum cycle pressure)
f) H/P water pressure control valve	SET (to mean cycle pressure)
g) cylinder water level control	CHECKED

5. Combustion

a) blower supply valve (at blower exit)	OPEN (10-20%)
b) air delivery valve (at control station)	CLOSED
c) blower switch	ON
d) multi-task control	STANDBY
e) fuel, ignition	ON
f) digital temperature readout	CHECKED
g) air delivery control	OPEN
h) multi-task control	MONITOR

Running Checks

1. Temperature of heater tubes.
2. Displacer motion
3. Gas pressures in engine
4. H/P water supply pressure
5. Gas spring control pistons
6. Cylinder water level control
7. Coolant flows
8. Lubricant pressures

Shut-Down Sequence

1. Combustion

a) fuel, oxygen torch valves	OFF
b) blower supply valve (at blower exit)	OPEN (100%)

2. Engine De-pressurization

a) working space supply valve	CLOSED
b) balance space vent valve	OPEN
c) gas spring vent valve	OPEN (as required to maintain displacer motion)
d) H/P water supply pressure control valve	OPEN
e) cylinder drain valves (bypass)	OPEN

3. Rotation

a) engine speed control	SET ('0')
b) dynamometer power	OFF

4. Reverse Preparation Sequence

a) fuel supply	OFF
b) nitrogen supply	OFF
c) instrumentation	OFF
d) water supply	OFF
e) electricity supply	OFF

N.B. Ensure cooling water and blower remain on for engine to cool sufficiently.